AUTOMOBILE ENGINEER

DESIGN · PRODUCTION · MATERIALS

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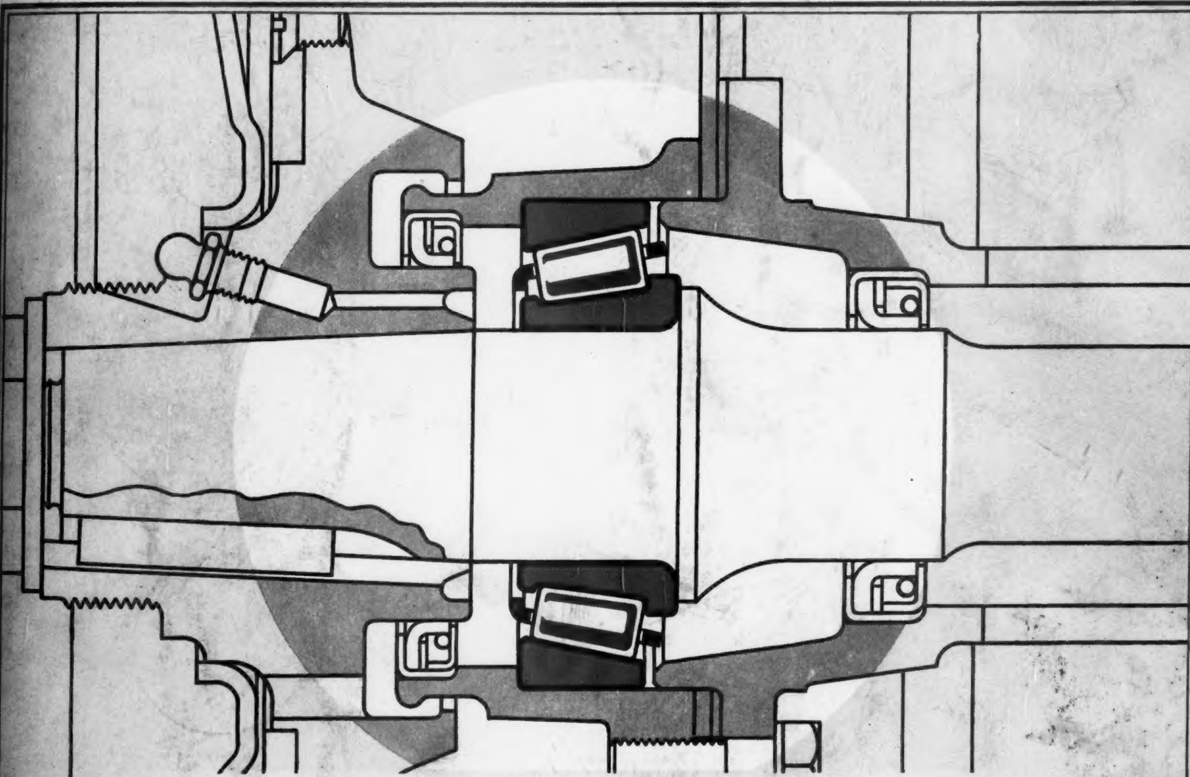

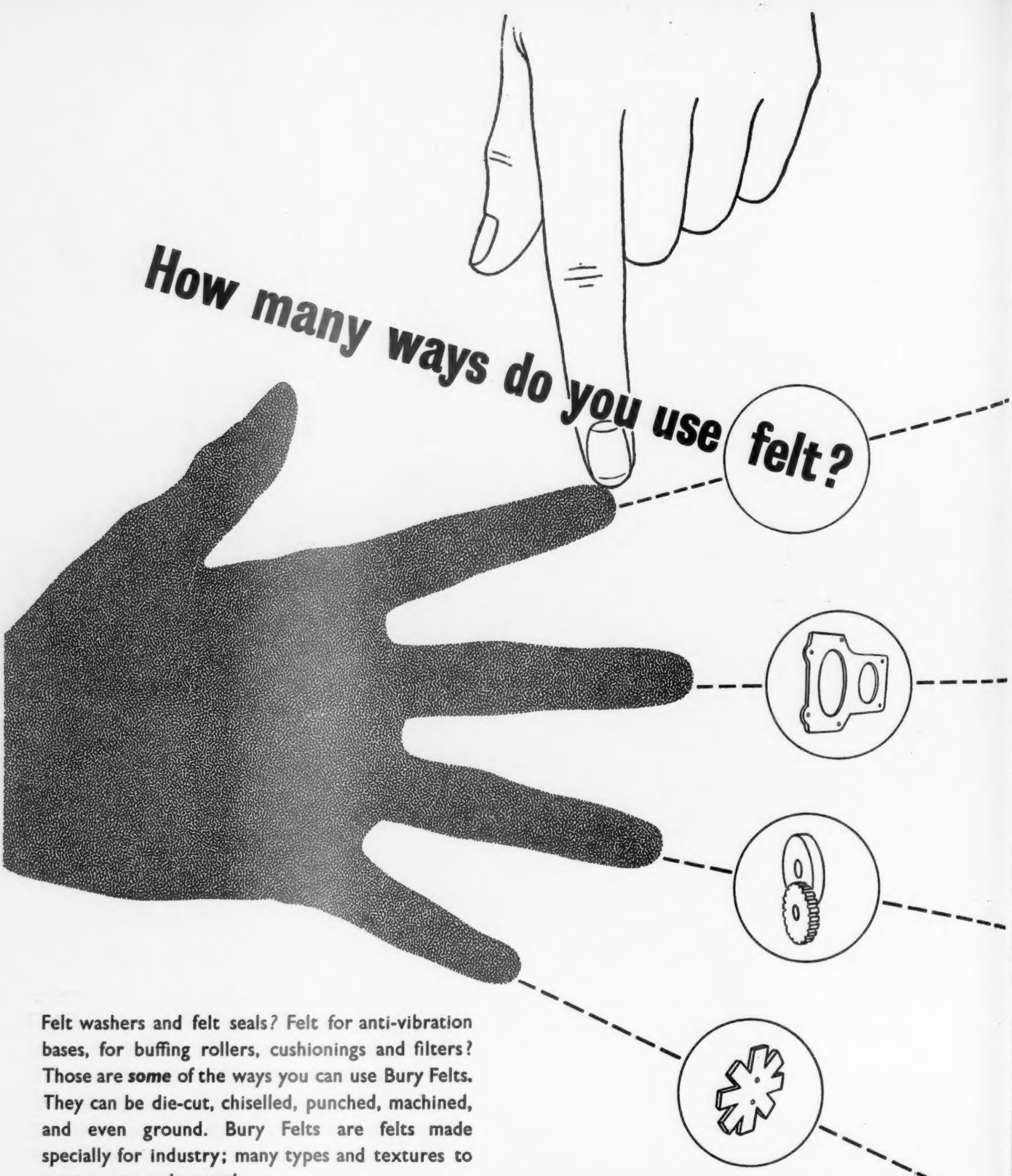

How many ways do you use felt?

AUTOMOBILE ENGINEER

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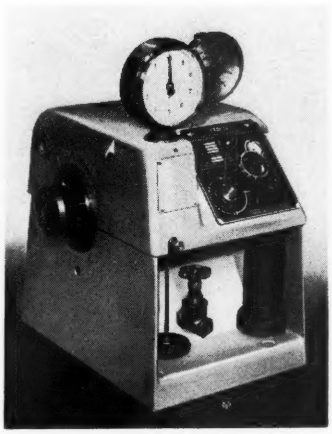

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DESIGN MATERIALS **AUTOMOBILE ENGINEER** PRODUCTION METHODS WORKS EQUIPMENT

Miniature Cars

ALMOST immediately following the advent of restrictions on fuel supply, there was a marked increase in the sales of very small cars with engines of less than 500 cm^3 swept volume—an increase greater, it seems, than was expected by even the most sanguine of the sponsors of these vehicles. This naturally has led the industry to speculate as to whether the foothold that this class of vehicle has gained is capable of being expanded, and if the potential demand is large enough to justify manufacture on a scale comparable with that of the production of more conventional private cars.

In the years immediately after the 1914-1918 war, a number of miniature cars, some with three and others with four wheels, were introduced. Of the three-wheelers, all except one soon went out of production. The model that continued to be a success was of pre-1914 origin: it was the most simple and one of the cheapest, if not the cheapest. None of the four-wheel cars was a great success.

Despite these failures, history was made when, in 1921, there was launched the now-famous Austin Seven, a small car designed to capture the then large motorcycle-and-sidecar market. From then on, it was found that as soon as a manufacturer gained a worthwhile following with a new model, a competitor recaptured the market with another, slightly larger but costing only a few pounds more; consequently, the small cars tended to become bigger as they were developed from model to model. A parallel to this experience is the fact that most customers, given the choice of an austerity and a de luxe version of the same car, prefer to pay the few pounds extra for the de luxe model.

The question that now arises is in what way do conditions differ from those before the 1939-1945 war, when so many small vehicles, and particularly the £100 car, failed to appeal to the public. In recent years, the density of traffic, especially in towns and cities, has been increasing rapidly, so manœuvrability and ease of parking are prime attractions. Taxation, as a whole, is now both considerably higher and much wider in scope, since it affects not only personal incomes, petrol and Road Fund licences, but also the purchase price of the vehicle and its equipment.

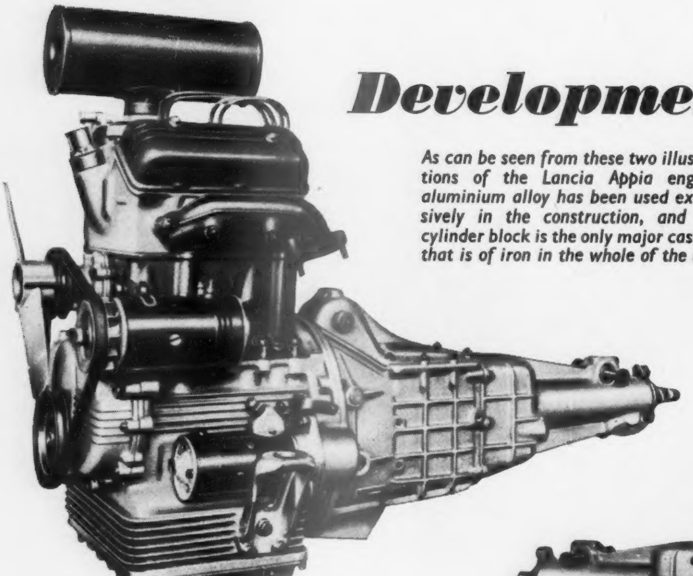
Mechanization is becoming more and more widespread in all aspects of life, as is demonstrated by the fact that many families now have equipment such as refrigerators, motor mowers, vacuum cleaners and washing machines. It is more than likely that this trend towards mechanization will extend still further, and miniature cars will be used increasingly for taking children to school and for shopping, particularly as the costs of both public transport and the delivery of merchandise are now so high. Cars of this type are becoming increasingly popular on the Continent and there is certainly a large market for them in many parts of the world.

At present, the demand is restricted because the prices of miniature cars that are available are not far enough below those of the cheapest of the more conventional small cars. However, large scale production of well designed vehicles by manufacturers with adequate resources might well change this. Since a miniature car inevitably is less comfortable than a more conventional vehicle of larger size, it can only be attractive if both its prime and running costs are very low. Undoubtedly the most economical type of vehicle, so far as both production and operation are concerned, is the three-wheeler. However, if the car is to have the widest possible appeal, it must be capable of carrying four adults, an arrangement difficult to obtain with the three-wheel layout. Optimum handling characteristics can only be obtained with four wheels, two steered and two driven, and with two tracks.

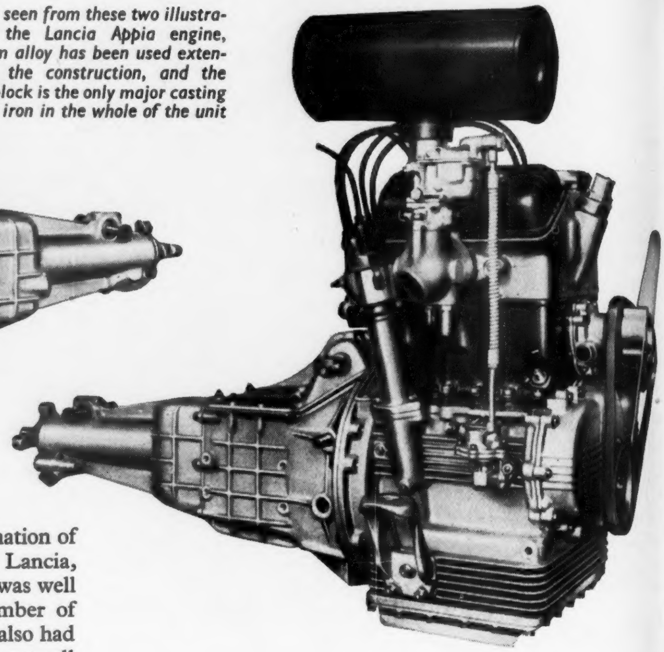
A number of trends and pressures that will affect the prospects of this type of vehicle some years hence are already apparent, but the time that it will take for these to have full effect cannot be readily determined. The standard of living in this and many other countries is constantly rising. This raises the question as to when it will reach the point where the miniature car is positively unacceptable. It must be borne in mind that the effect of this rising standard is to a certain extent offset by the fact that availability of an ever-widening range of consumer goods, such as television and other amenities, tends to absorb increasing portions of personal incomes.

Public transport organizations are pressing for the exclusion of all except business and through-traffic from large towns and cities. This is a political hazard that might at any time seriously affect the sales of vehicles designed largely for shopping and other local runs. It is even possible, but admittedly not probable at present, that the tax on motor fuels might be drastically reduced, with far-reaching effects on motor vehicle design in general. The advent of atomic power might lead to the discovery of some inexpensive way of synthesizing liquid fuels, or some unforeseen development might even revolutionize power units for motor vehicles. Fortunately, most of these potential developments are unlikely to come so quickly that manufacturers would be caught with unamortized tools on their hands; therefore, at least in these respects, the long-term risks are not likely to be a serious deterrent.

Development of the Lanci



As can be seen from these two illustrations of the Lancia Appia engine, aluminium alloy has been used extensively in the construction, and the cylinder block is the only major casting that is of iron in the whole of the unit



LAST year was the fiftieth anniversary of the formation of Lancia & C. Before he founded the firm, Signor Lancia, was a racing driver and tester employed by Fiat and was well known in the years 1902-1905 as winner of a number of prizes in international racing and competitions. He also had latent talent as an engineer, and in 1906 started in a small way by employing about a hundred people. The original factory was on one corner of the site currently occupied by the Lancia works.

The firm was among the first to equip cars with electric lighting: this was in 1913. Also, Lancia was well to the fore with chassisless construction: the Lambda, which was first produced in 1921, was of unitary construction, that is, the frame was integrated with the body. Another outstanding feature of this vehicle was the independent suspension system, which was similar to that employed on the modern cars made by Lancia. The next model, the Augusta, which was developed in 1931-1932, was of fully chassisless construction.

An acute angle Vee layout was first adopted for Lancia engines forty years ago. Its main advantage is compactness. The fact that the arrangement has withstood the test of time shows that it is sound in conception. It is of interest to trace the later stages in the development of this layout. Before the last war, it was employed in an engine of 1,352 cm^3 capacity. This unit had a single overhead camshaft and three rocker shafts, one above and one on each side of the central overhead camshaft.

The axes of the valves were inclined to form a Vee in a transverse vertical plane above the axis of each cylinder. Those inclined towards the centre of the engine passed under the camshaft and were actuated by rockers of bell-crank form, pivoted on the central shaft. One arm of the bell-crank actuated a tappet immediately above the end of the valve stem and the other arm extended down on the other side of the rocker shaft and had the cam follower formed on its end. The outwardly inclined valves were each actuated by two rocker levers. One lever was pivoted on the upper shaft; it was simply a pendant arm with a cam follower on its lower end and a short projection extending sideways from a point near its pivoted end. This projection actuated the second rocker, which was of conventional form, pivoted on the outer shaft. The other end of this second rocker bore directly on the valve stem. As can be seen from the accompanying illustration, the sparking plugs were in a relatively inaccessible position. Other features of this engine were a deep skirted cylinder block and crankcase unit, of cast iron, with wet liners. The cylinder head was also of cast iron, but the sump was of cast aluminium.

In November, 1939, the Ardea engine was introduced. This had a capacity of 903 cm^3 and, mechanically, it differed from the earlier unit in that the wetted length of the liners was greater and the crankcase was virtually a separate casting of aluminium alloy bolted to a flange an inch or so below the base of the cylinder block jacket. In this engine, the main bearings were carried by webs formed integrally with the cylinder block, and were totally enclosed by the crankcase casting.

The valve gear was simplified, although the overhead camshaft was retained. Vertical tappets carried in a block bolted on top of the camshaft cell actuated the rockers. Each valve had a separate rocker assembly. The two arms of the rocker were arranged one at each end of a short spindle. Between these arms, a bearing cap was fitted over the spindle to retain it in its housing in a block cast integrally on the cylinder head. The valves of each cylinder were inclined to form a Vee in a plane set, not transversely as in the earlier engine, but diagonally at an angle between the longitudinal and transverse axes of the engine. This was to enable the plug boss to be incorporated at the side of the head, and thus to improve accessibility to the sparking plugs. The arrangement of valves in this way was possible only because the spacing between the cylinder axes was relatively large by comparison with that of conventional in-line units.

The latest engine, the Lancia Appia, is a logical development from the two earlier ones. When this engine was designed, the manufacturers had four main aims. The first was at simplification to reduce cost, the second at compactness

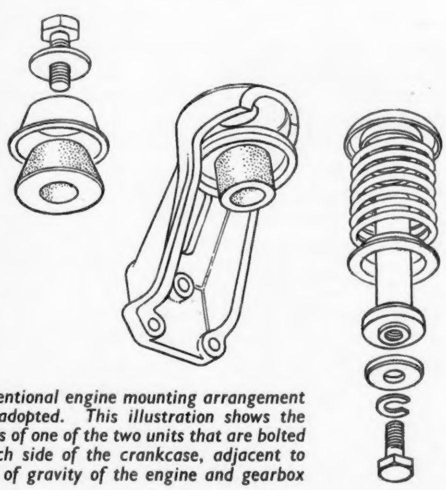
Appia Engine

An Acute Angle Vee Layout That is Noteworthy for its Compactness and Light Weight

to enable a larger proportion of the space in the car to be made available to the passengers, the third aim was at ease of production, and the fourth at light weight to improve economy, both as regards prime and running costs. To reduce the weight, the amount of cast iron used has been minimized by the employment of a separate cylinder block, while the crankcase, which carries the main bearings, is of cast aluminium.

Although this engine is lighter than its predecessors, the advantage gained in this respect, by comparison with in-line engines, is not so marked as might at first be supposed. This is owing to the relatively large volume enclosed by the cylinder block, which is almost square in shape. Because of this symmetrical layout and the consequent rigidity of the block, and also because of the symmetrical positioning of the studs, it has not been necessary to employ separate liners to avoid bore distortion.

To simplify servicing, the overhead camshaft layout has been abandoned and twin camshafts are now installed, one each side of the crankshaft. Although this has not greatly simplified the valve actuating gear arrangement, it has a number of advantages. One is that the valve timing is not disturbed when the head is removed. The timing drive is shorter and simpler, and the sparking plugs are favourably positioned to give good combustion characteristics.



An unconventional engine mounting arrangement has been adopted. This illustration shows the components of one of the two units that are bolted one on each side of the crankcase, adjacent to the centre of gravity of the engine and gearbox

Compactness has been obtained by the employment of only two main journal bearings, whereas on the earlier engines there were three. One of the bearings is carried in the front wall and the other in the rear wall of the aluminium casting that forms the crankcase. A transverse rib, mid-way between the ends of the crankcase carries the intermediate bearings of the camshafts and stiffens the casting. There is a hole in this rib to clear the crank web between the pins for numbers 2 and 3 cylinders. The adoption of the two- instead of the three-bearing layout, in conjunction with the separate aluminium crankcase, was undoubtedly a wise move, since the alignment of the bearing housings presents less difficulty, and it is easier to distribute crankshaft loading evenly between two bearings than between three. Because of the flexibility of the crankcase, the alignment of the front and rear bearings is virtually dependent on the trueness of the joint face at the base of the cylinder block.

A stroke:bore ratio of 1·103:1 has been adopted. This gives a bore that is large enough for the accommodation of valve ports of adequate size in the hemispherical combustion chambers, and there is ample space for the two main journal bearings. There would have been no point in choosing a ratio giving more nearly square dimensions, since this would have increased unnecessarily the overall length of the unit. A high connecting rod length:stroke ratio, of 2·135:1, has been adopted because the axes of the cylinders intersect at a point a long way below the crankshaft axis; in fact, this point is two or three centimetres below the base of the sump. This arrangement is regarded as a good compromise in the interests of compactness. The angle of the connecting rod axis relative to the cylinder axis, when the piston is at top dead centre, is 5 deg, and this increases the angularity of the rod under thrust conditions; nevertheless, the rod is long enough to avoid the necessity for incorporating a slot in the skirt of the cylinder to clear it. The joint face between the crankcase and cylinder block is 10·1 cm (3·98 in) above the axis of the crankshaft.

At the engine speed at which maximum b.h.p. is obtained, the piston speed is 2,365 ft/min. The ratio of maximum torque to the torque developed at maximum b.h.p. is 1·28:1, and the speed ratio for these two conditions is 0·555:1. In terms of b.h.p/in^2 piston area, a figure of 1·7 is obtained and the figure for the b.h.p/litre is 34·9. The dry weight of the engine, complete with flywheel, is 93 kg (205 lb). Thus, the b.h.p. developed per pound is 0·185. A minimum brake specific fuel consumption of 222 gm/CV-hr (0·49 lb/b.h.p-hr) has been obtained.

So far as overall dimensions are concerned, the height of the unit without the air filter is 24$\frac{3}{4}$ in, its width is 18 in, and the overall length, from the front of the fan hub to the rear face of the flywheel housing is 15 in. The engine is installed at an angle of 1 deg 32 min 24 sec from the horizontal. Its rear mounting is a rubber bush, the axis of which is horizontal in a transverse plane, in an eye on top of the gearbox extension.

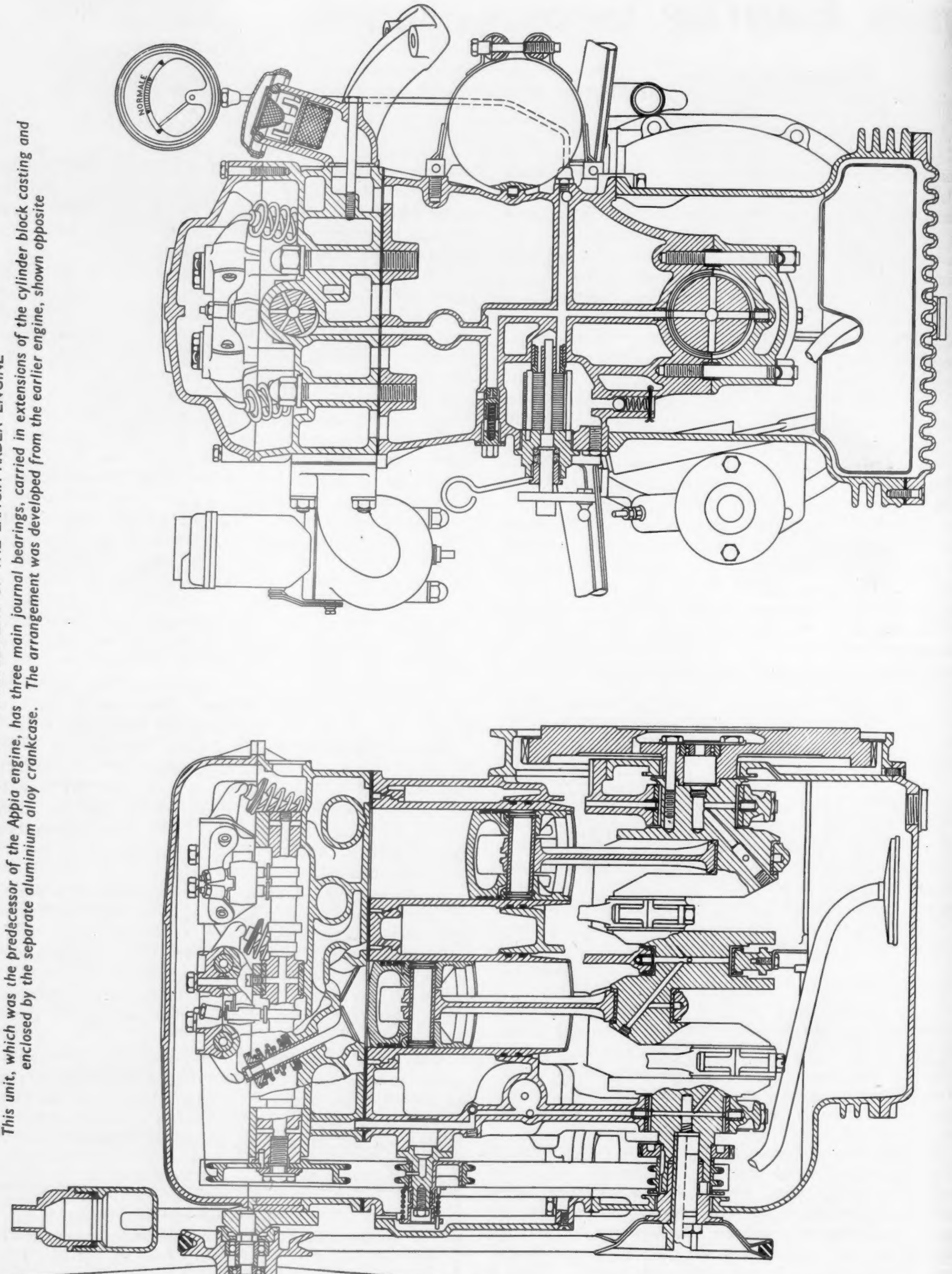
The front mounting is unconventional. A bracket is bolted to each side of the crankcase at a point near the centre of gravity of the engine. Formed at the upper end of this bracket is a horizontal platform with a hole in its centre. A shouldered circular rubber, also with a hole in its centre,

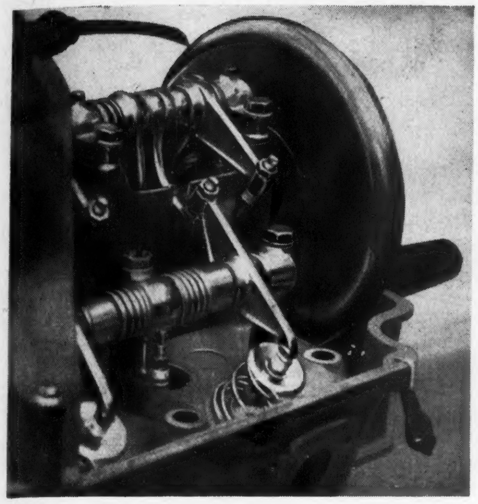
SPECIFICATION

Four cylinders. Bore and stroke 68 mm and 75 mm respectively. Swept volume 2,483 cm^3. Maximum b.h.p. 38 at 4,800 r.p.m. Maximum torque 51·96 lb-ft at 3,000 r.p.m. Compression ratio 7·4:1. Forged, two-bearing crankshaft. Overhead valves, operated by a camshaft on each side of the crankcase. Solex downdraught carburettor, type 32/30 BI. Approximately hemi-spherical combustion chamber, with a quench shoulder on the side remote from the plug. Mechanical fuel lift pump.

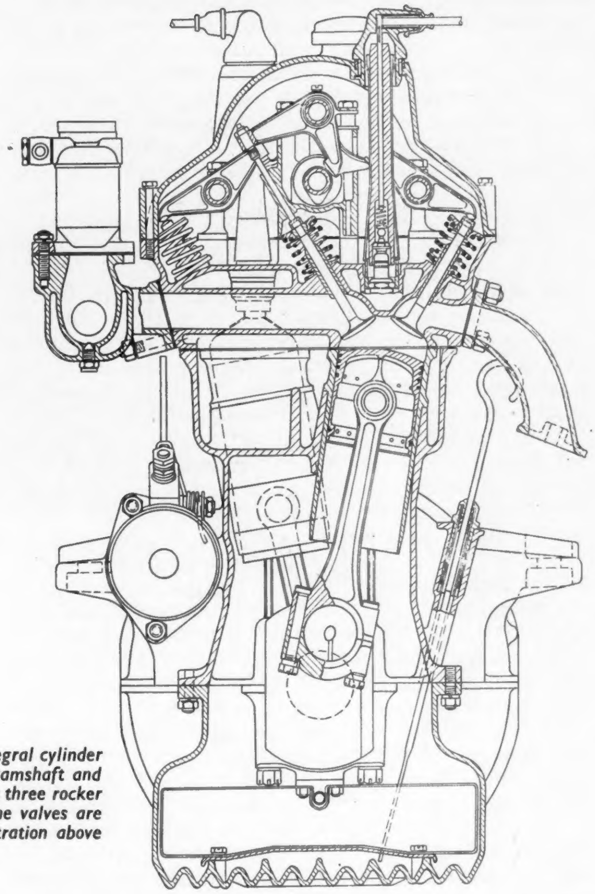
SKETCH SHOWING THE ARRANGEMENT OF THE LANCIA ARDEA ENGINE

This unit, which was the predecessor of the Appia engine, has three main journal bearings, carried in extensions of the cylinder block casting and enclosed by the separate aluminium alloy crankcase. The arrangement was developed from the earlier engine, shown opposite





Valve gear on a demonstration model of the engine illustrated on the right, which was at the 1936 Paris Show



The 1,352 cm³ engine, which was introduced before the Ardea, has an integral cylinder block and crankcase. Unlike the Ardea, which has a single overhead camshaft and tappets actuating separate rocker assemblies for each valve, this unit has three rocker shafts, one above and the others one on each side of the camshaft. The valves are actuated by the complex system of rocker levers, also shown in the illustration above

is assembled from below and spigoted into the hole in the bracket. A dished washer, which forms the seat for the upper end of a coil spring, is fitted under this rubber and there is another seating washer at the lower end of the spring. A boss formed centrally under the rubber extends into the coil spring to form an auxiliary spring and bump stop. Above the platform on the bracket is a conical rubber in a cupped pressing. Both the rubber and the pressing have a hole in the centre, and the whole assembly is held together by a headed pin inserted from below, through the spring, its seating washers, the rubbers above and below the bracket, and the cupped pressing. This pin is retained in the mounting assembly by a set bolt screwed into an axial hole in its upper end; at its lower end it is spigoted into a hole in a bracket on the vehicle structure, to which it is secured in a similar manner by another set screw.

Cylinder block

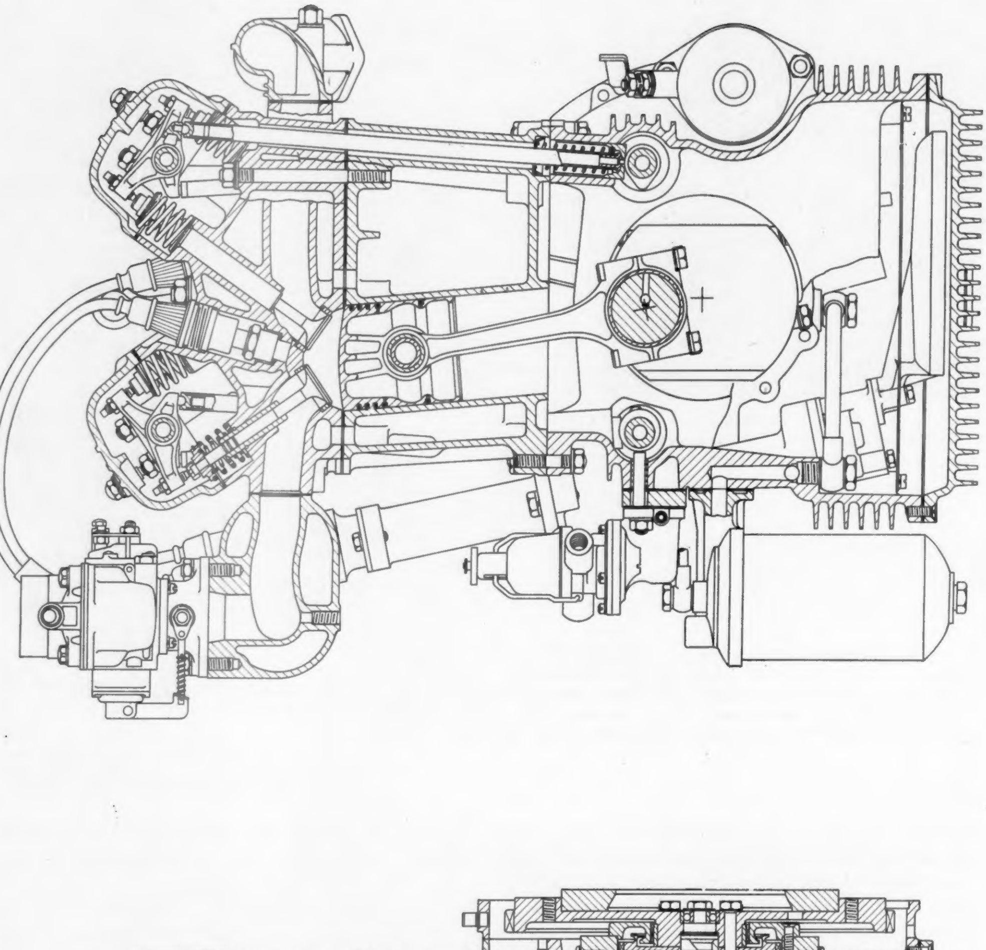
The cylinder block is the only major casting made of iron in the whole of the engine and gearbox assembly. Although the design is compact so far as overall length is concerned, it is wider than more conventional in-line designs of cylinder block. However, the width is no greater than is necessary in any case to accommodate the rotating crankshaft, and the almost rectangular plan-form is a good feature so far as rigidity is concerned. The overall dimensions of the casting are: height 132 mm (5 $\frac{3}{16}$ in), length 254 mm (10 in), and width 264 mm (10 $\frac{25}{64}$ in). Apart from the fact that cylinder bore distortion is unlikely to occur in a block of such symmetrical layout, there is a good reason for abandoning the use of wet liners. In the earlier designs, in which liners of this type were employed, the main journal bearings were carried in extensions of the cylinder block casting; but in this engine, in which the bearings are in the separate crankcase casting, the rigidity of the crankcase and cylinder block assembly is dependent mainly on the stiffness of the block, which would be reduced by the employment of wet liners.

As has already been mentioned, an acute angle Vee-type layout has been adopted. The included angle is 10 deg 14 min. Numbers 1 and 3 cylinders are on the left-hand side, as viewed from the rear, and numbers 2 and 4 are on the right.

Between each pair of cylinders, the water jacket space is 6 mm ($\frac{15}{64}$ in). The cylinder walls are 4 mm ($\frac{5}{32}$ in) thick, as also are the side and end walls of the jacket. All the machined faces on each end and on the top and bottom of the block respectively are in one plane. There is no machining on the sides of the block.

The twelve 8 mm and one 10 mm diameter cylinder head holding-down studs are shouldered to form a water seal where they are screwed into open bosses in the upper decking of the block. Had blind bosses been employed, they would inevitably have been larger so the water flow would have been unnecessarily obstructed. As an added precaution against the leakage of water, domed nuts are used, in conjunction with plain washers, to hold down the head. At the base of the block, there is a flange on each side to carry the studs that secure it to the crankcase casting. The front face is machined locally to form the joint face for the upper edge of the timing drive cover and also to receive the water pump. A small bridge-piece is bolted to a face machined on the rear of the block to close the top of the flywheel housing. An aperture in this bridge-piece forms the timing window; it is closed by a steel plate.

To effect the connection between the cylinder block and crankcase casting, six 8 mm diameter studs are employed on the left-hand side and five on the right-hand side. The

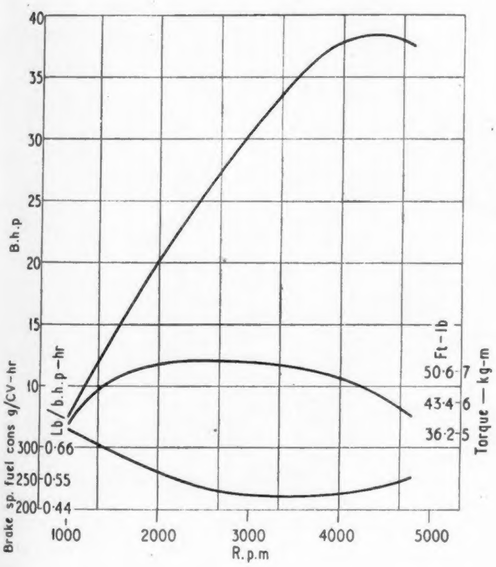


ARRANGEMENT OF THE LANCIA APPIA ENGINE
Bore 68 mm. Stroke 75 mm. Swept volume 2,483 cm³

reason why one has been omitted on the right-hand side is to provide clearance for the distributor drive. There are two more studs at the rear. Location is effected by 7 mm and 12 mm diameter dowels, one in the front and the other in the rear joint face.

Crankcase

A rectangular casting of aluminium-copper-silicon alloy forms the crankcase. Externally, it is finned on each side and at the front. Internally, there is a transverse rib mid-way between its ends, the function of this rib being to carry the intermediate bearings of the two camshafts and to stiffen the side walls of the casting. There is no joint washer between the machined face on the flanged upper edge of the crankcase casting and the cylinder block. The lower edge of the crankcase is also flanged and machined, to form a joint face for the cast aluminium alloy, dished base plate. This base plate is fitted with a 0·4 mm thick, Vellumoid joint washer and is finned externally. The choice of material for this joint washer is important because it is below the oil level in the sump. Twenty-two, 6 mm diameter countersunk set screws secure the base plate to the crankcase. Locking is effected simply by the grip of the conical seatings of the heads of the set screws in the countersinks. Slightly more than 1½ cm (⅝ in) above the joint face at the base of the crankcase, that is, well below the oil level, a horizontal baffle plate is fitted in the sump. It is secured by set screws to four bosses, one in each corner of the casting.



Above: Performance curves of the engine complete with fan, air filter, dynamo and exhaust manifold. The results are corrected to standard atmospheric conditions

At the front, the machined faces include the joint face for the timing gear cover, and the forward ends of the bosses that carry the housings for the front main journal and the front bearings of the two camshafts. All these bearing housings are of cast iron, and are flanged and spigoted, from the front, into holes in the bosses in the crankcase wall. Each camshaft bearing housing is secured to the boss in the crankcase by three countersunk set screws and it carries a BS PB20 bush.

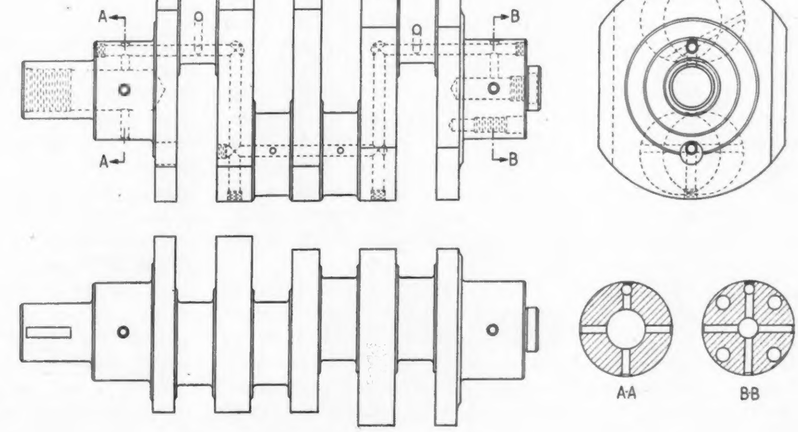
The front main journal bearing housing is secured by five set bolts locked by tab washers. It is 78·000 mm to 77·987 mm outside diameter and the hole in which it fits is 77·990 mm to 78·009 mm diameter, so the clearance between the two varies from −0·010 mm to +0·022 mm. The bearing is not split, since it can be assembled over the front end of the shaft. It is a Vandervell bush, 65 mm outside diameter by 55 mm inside diameter by 32 mm long, and is of a 50/50 copper-lead alloy, with a 0·7 mm thick copper-lead-indium lining on the bearing surface. Location is effected by a 6·5 mm diameter dowel inserted radially through the spigot portion of the housing into a shouldered hole in the bush. The reason why the hole in the bush is shouldered is, of course, to prevent the dowel from contacting the surface of the journal and scoring it.

The rear journal bearing is of the same material, but its front end is flanged to form a thrust face to bear against the adjacent crank web. This bearing is 70 mm (2·75 in) outside diameter by 55 mm (2·16 in) inside diameter; both its end faces, as well as its bore, are copper-lead-indium lined. This bush also is dowel-located. It is thicker than that at the front because it locates the shaft axially. Thin shell type main journal bearings could not be employed because they have not the degree of conformability necessary to accommodate the crankshaft deflections likely to be experienced with this two-bearing layout.

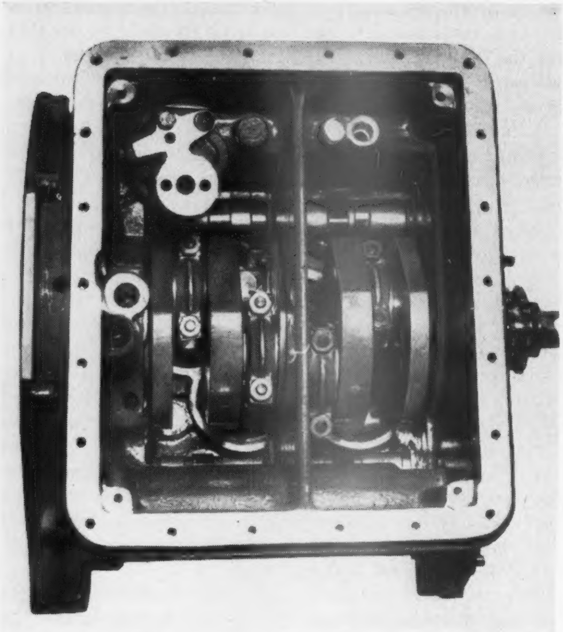
This bearing is housed in a flanged iron casting, assembled from the rear and secured by ten studs screwed into the crankcase. A joint washer of Vellumoid is interposed between the flange and the crankcase. The rear camshaft bearing bushes are each located by a 6 mm diameter dowel, and large diameter plugs are screwed into the rear ends of their bosses to prevent oil from escaping into the flywheel housing, which is formed in the rear end of the crankcase.

Crankshaft, connecting rod and piston assembly

The crankshaft is a forged nickel-chromium-molybdenum steel (17 NCD Tmp) component, and is hardened. Although it closely resembles the conventional two-bearing four-throw type, the crank pins are not all in the same plane.



Right: Although a four-throw crankshaft is employed, it is unconventional in that the front pair of crank pins is offset to one side of their webs, while the rear pair is offset the same amount to the opposite side



Above: Crankshaft assembly viewed with the cylinder block removed. Only two main journals are employed, and the length of the crankshaft, between the outer faces of the front and rear webs is less than 7 in

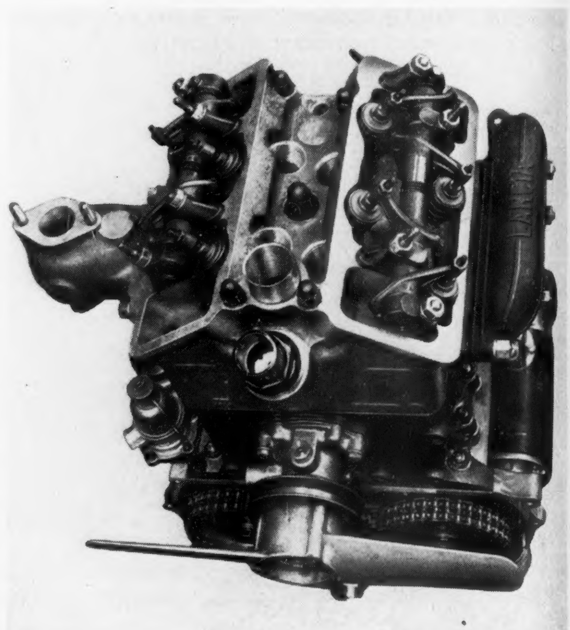
This unusual feature has been adopted, of course, because of the acute angle V-arrangement of the cylinders. All five webs are similar in shape: with numbers 2 and 3 cylinders, for example, at top dead centre, the shape of each web, as viewed from the front, is that of a circle with flats machined vertically on each side. The webs are wider than is usual on a conventional four-throw shaft, and the pins are offset to one side or the other of their vertical axes. Thus, with the crankshaft in this position, the pin for number 1 cylinder, which is at bottom dead centre, is offset to the left, as also is that for number 2 cylinder, which serves the other bank of cylinders and is at top dead centre. Similarly, the pins for numbers 3 and 4 cylinders are offset to the right. Because of these offsets, balance weights are necessary to obviate the rotating couple that otherwise would be experienced. These weights are formed, integrally with the webs, by taking off less material to form the flats on the appropriate sides of numbers 1 and 2 and numbers 4 and 5 webs of the crankshaft.

The front and rear webs are about 13 mm ($\frac{1}{2}$ in) thick, the centre one is 16 mm ($\frac{5}{8}$ in) thick and the other two are 20 mm ($\frac{25}{32}$ in) thick. All the crank pins are 21 mm long and the overall length of the shaft between the outer faces of the front and rear webs is only 176 mm ($6\frac{13}{16}$ in). The centre web is 92·5 mm ($3\frac{11}{16}$ in) wide, while the width of the others is 100 mm ($3\frac{13}{16}$ in). When two of the pistons are at top dead centre, they are, of course, always in opposite banks of cylinders; the included angle between the major axes of their connecting rods is 21 deg. The point of intersection of these axes is on the longitudinal axis of the crankshaft main journal bearings.

At the rear of the shaft, the flywheel and the oil sealing arrangements adopted are unconventional. The flywheel is a disc of chromium-manganese steel (35 CM4 Bon) on the periphery of which are machined the 104 starter gear teeth. This ring gear meshes with an eight-tooth pinion. An 185·9 mm (7.2 in) diameter cast iron ring is spigoted into the rear face of the flywheel to locate the clutch and to form the thrust face, against which the driven-plate assembly is pressed when the clutch is engaged. This ring is shrunk-in. A ball bearing for the front end of the clutch shaft is carried in a 26 mm diameter housing in the centre of the flywheel. The overall diameter of the flywheel is 259·5 mm.

Radial location of the flywheel is effected by a 24 mm diameter spigot extension of the crankshaft, to which the component is secured by five 8 mm diameter set bolts. Between the flywheel and a shoulder on the shaft, is an oil return ring with helical teeth machined on its periphery. The helix angle is 45 deg and is set in such a manner that, as the ring rotates in a counterbore in the rear end of the barrel type housing for the rear main journal bearing, any oil that tends to leak between the teeth is returned to the crankcase. A collar is formed on the rear face of the toothed ring; it is lipped to form an oil thrower to prevent any oil that may leak past the teeth from going radially inwards and thence into the housing for the single-dry-plate clutch.

Right: On the Lancia Appia, one of the rocker shafts is shorter than the other; only the centre pair of rockers on each shaft have coil springs interposed between them, all the other rockers being retained between the front and rear pairs of pedestals, which are bolted to the head



A cover plate, spigoted into the counterbore in which the ring rotates, encloses the whole assembly. The joint between the cover plate and the main journal bearing housing is made good with a sealing washer. A lipped collar round the centre of the cover plate projects forwards and is surrounded by the collar on the rear face of the oil return ring. Another helical gear, carried on the flywheel boss, rotates in the bore of this collar on the cover plate to form an additional seal.

Connecting rods of nickel-chromium-molybdenum steel (38 NCD4 Bon) are employed. They are of H-section and their centre-to-centre length is 160 mm (6·3 in). The minimum cross sectional dimensions of each rod are $\frac{9}{16}$ in by $\frac{7}{16}$ in over the flanges. In each big end, a divided bearing of copper-lead, lined with babbit is employed. The bearing is flanged at each end, and is located against rotation by the bolts that secure the bearing cap. The hole for the bolt on

each side breaks into the bore in which the bearing is carried, and the shank of the bolt registers in a groove machined vertically in the outer periphery of the bearing, adjacent to the abutting faces of the halves. The crank pin diameter is 46 mm (1·48 in) and the bearing length is 22 mm (1$\frac{3}{8}$ in). There is a diametral clearance of 0·009 mm to 0·035 mm between the bearing and the pin. A single rib round the periphery of the cap provides the stiffness necessary to ensure that the cap and bearing assembly does not distort under load.

A 22 mm ($\frac{7}{8}$ in) long phosphor bronze bush is pressed into the small end to carry the gudgeon pin, which is of plain cylindrical section. The outside diameter of the gudgeon pin is 20 mm (0·787 in), its inside diameter is 15 mm (0·59 in) and it is made of hardened nickel-chromium steel (15 NC5 Cmt). Axial location is effected in the conventional manner by Seeger type circlips in grooves in the ends of the cross holes in the piston. The bearing length of the pin in each cross hole is 17 mm ($\frac{5}{8}$ in).

The low expansion aluminium alloy pistons have slightly domed crowns, to obtain the degree of compression required, and are of the solid skirt type. This type is best from the point of view of mechanical strength, but most manufacturers prefer the slotted skirt piston because it does not have to be manufactured to such close tolerances. Possibly, the solid skirt type has been adopted for this engine because of the relatively large thrust angle due to the inclination of the connecting rods relative to the cylinder axes. The piston crowns are horizontal, that is, parallel to the top deck of the cylinder block, each being at an angle of 5 deg 7 min to the plane normal to its cylinder axis. Inside the piston, three arched ribs, parallel to the gudgeon pin axis, transfer the loads from the crown to the piston bosses.

There are three compression rings and two oil control rings on each piston. All three compression rings and one of the oil control rings are above the gudgeon pin, while the other oil control ring is below it. The compression rings are of plain rectangular section and the top one is chromium plated. Each has a face width of 2 mm. Both the oil control rings are of the slotted type. An unusual feature of the arrangement is that ducts from the groove for the upper ring are drilled to break out in the cross holes for the gudgeon pin to afford positive lubrication for these bearing surfaces.

Timing drive and valve gear

The timing drive is enclosed by a cast aluminium alloy cover bolted, together with a joint washer, to the front face of the crankcase. Assembled on to the 32 mm diameter front extension of the crankshaft are: a short distance piece, the nickel-chromium-molybdenum steel (38 NCD4 Bon) sprocket for the three-strand timing chain, an oil thrower ring and the cast pulley for the V-belt. A lip type oil seal housed in the timing cover bears on the boss of the pulley. The dished thrower ring is clamped between the pulley and the sprocket and it enshrouds the seal housing, which is lipped to prevent oil that drains down the front wall of the cover from running directly on to the seal. The axes of the two camshafts are 40·72 mm (1·64 in) above and 87 mm (3·42 in) each side of the axis of the crankshaft. The three-strand timing chain is passed round the two camshaft sprockets, between which the crankshaft sprocket meshes with the lower face while the eccentric jockey sprocket meshes with the upper face of the chain.

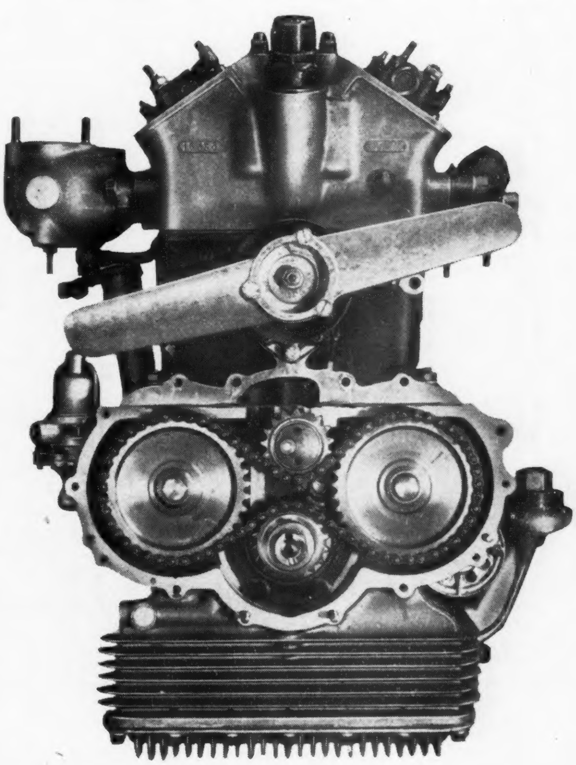
An interesting feature of the eccentric jockey sprocket is that it is held against the chain by a small hydraulic jack actuated by the oil pressure of the lubrication system. The sprocket and eccentric assembly is carried on a 12 mm diameter spindle, which is reduced to 10 mm diameter and threaded at its rear end, where it is passed through a hole in a lug projecting upwards from the main journal bearing housing. A retaining nut is fitted on this threaded end. An actuating lever is secured by countersunk set screws to the rear face of the eccentric. This lever extends horizontally over a vertical cylinder formed in the main journal bearing housing, and a spring-loaded plunger in the cylinder bears against its end. The eccentric is retained by a Seeger type circlip on the front end of the spindle.

When the engine is stationary, only the spring acts on the plunger and lever and maintains the sprocket in contact with the chain, but when the engine is started, oil pressure in the cylinder assists the spring. A white metal lined, bronze bush is pressed into the bore of the sprocket, which is free to float axially on the eccentric and thus to align itself with the chain. A small lug secured to the front wall of the crankcase extends over the crankshaft sprocket. Its function is to prevent the chain from slipping off if the throttle is opened suddenly before the oil pressure has had time to build up and cause the jockey sprocket firmly to take up the slack.

The camshaft sprockets are of nickel-chromium steel (35 NC9 Bon). Each is secured by a single, 10 mm diameter bolt screwed into an axial hole in the front end of the shaft. There are eleven more holes round this central hole in the shaft, while in the wheel there are twelve holes on a pitch circle of the same diameter. When the timing has been set, the wheel is located relative to the shaft by a dowel inserted into the pair of holes that happens to be in line. This dowel is retained by the head of the central set bolt, which is large enough to blank off the forward ends of the holes in the sprocket.

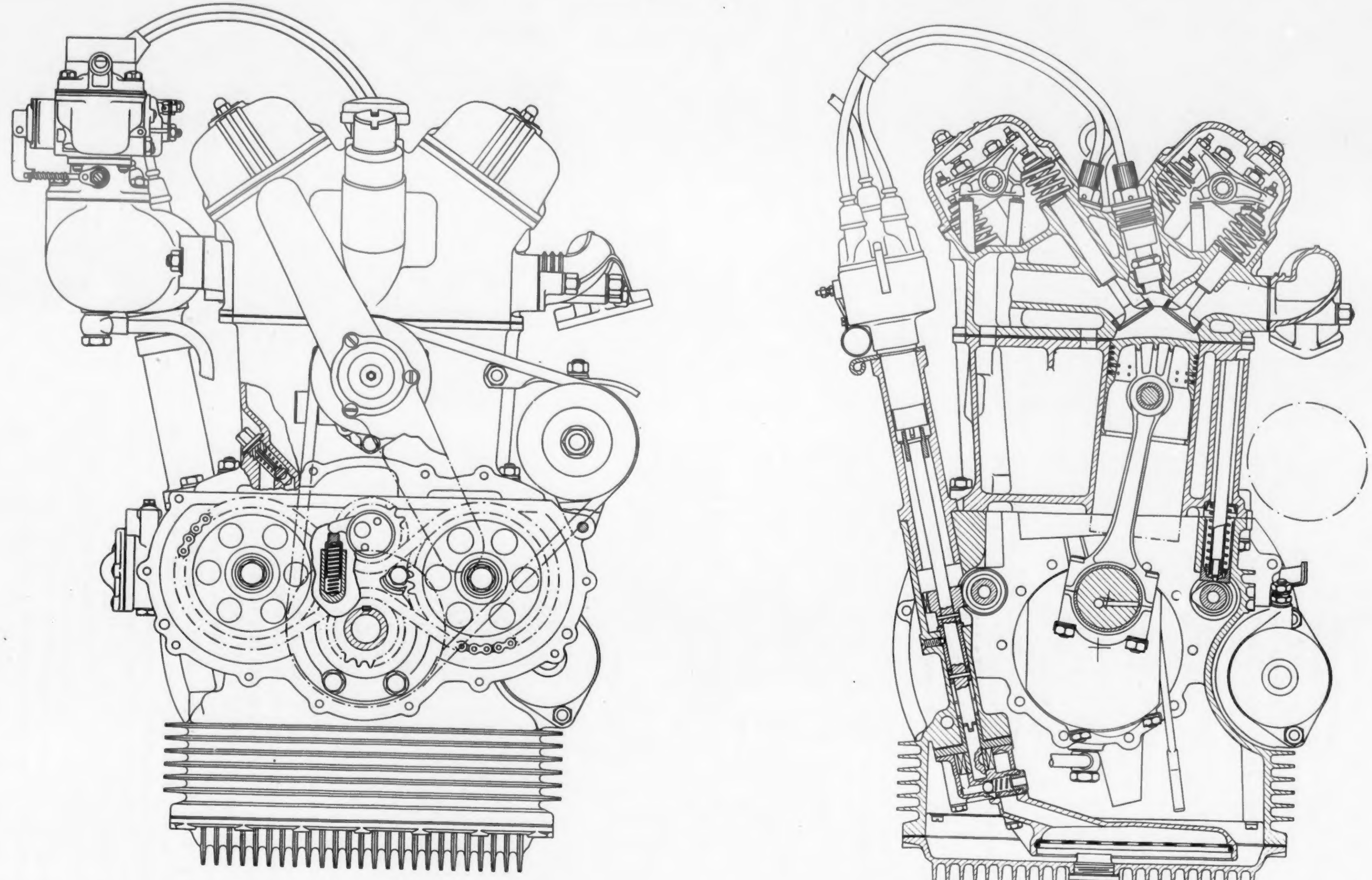
Axial location of the camshaft is effected by the front bearing, which is between the sprocket and a collar round the shaft. The front bearing bush is flanged at its forward end and pressed into a flanged cast iron housing. This housing is spigoted into a hole in the front wall of the crankcase, to which it is secured by three bolts. The

The Appia has two camshafts, one on each side of the crankcase. This illustration shows the engine with the front cover removed to disclose the unconventional timing drive and chain-tensioner arrangement



CROSS SECTIONS OF THE LANCIA APPIA ENGINE

The illustration on the left shows the timing drive layout and the hydraulic chain tensioning device. That on the right is a cross section through the oil pump and distributor drive, and also shows the arrangement of the valve actuating gear with its spring-loaded tappets



other two bearings are fitted one in the intermediate web of the crankcase and one in the rear wall. They are not carried in cast iron housings, as is the front bearing, but instead are pressed directly into their bosses. The front bearing is 21 mm long, the length of the centre one is 20 mm, and that of the rear bearing is 22 mm. Their diameters are respectively 32 mm, 35·5 mm and 20 mm.

The maximum positive acceleration of the tappet, on the flank, is 1,635 m/sec^2 and the maximum negative acceleration, on the nose of the cam, is 712 m/sec^2. A lift of 4·7 mm has been adopted, and the nominal period of the cam is 128 deg. Between the cams, the shaft is 20 mm outside diameter and 10 mm inside diameter. Machined on the right-hand camshaft, are an eccentric to actuate the fuel pump and a spiral gear to drive the oil pump and distributor. The eccentric is between the front and intermediate bearings, and the spiral gear is between the intermediate and rear bearings.

Every effort has been made to keep the weight of the reciprocating components of the valve actuating gear to a minimum. Case hardened steel tappets are carried directly in 22 mm diameter holes in bosses on the aluminium alloy crankcase. They are of unusually light section, their walls being about 1 mm thick. Each is held down against the cams by a coil spring, the upper end of which seats in a dished washer in a counterbore in the lower end of the hole in the cylinder block, through which the push rod is passed. The function of this spring, of course, is to reduce the load on the valve springs on the cylinder head.

Tubular push rods are employed. They are 10 mm outside diameter and 8 mm inside diameter by about 288 mm long, and they have hardened steel end fittings. The spherical lower end seats in the tappet and the upper end is cupped to receive a small, hardened steel, spherical ball-pin, which is spigoted into a hole in the end of the cast iron rocker. A screw type tappet adjuster is fitted in the other end of the rocker and bears on a hardened steel cap fitted over the end of the valve stem.

Each case hardened steel rocker shaft is 16 mm outside diameter by 10 mm inside diameter, and is carried by four aluminium alloy pedestals. The left-hand rocker shaft actuates the exhaust valves and is longer than the right-hand one, which actuates the inlet valves. Axial location of each is effected by a set screw, inserted radially into the rear pedestal. The front and rear rockers are each located between two pedestals, while two distance tubes with a coil spring between them are interposed between the other two rockers to constrain them against the intermediate pair of pedestals.

Details of the valves and springs are given in the accompanying Table. Each valve has two springs, the lower ends of which seat on a flat washer round the valve guide and the upper ends of which bear against a shouldered washer retained by split tapered collets in a groove round the valve stem. The housing for the exhaust valve guide is about 72 mm (3⅞ in) long and is completely surrounded by the water jacket. On the other hand, the inlet valve guide housing is only 24 mm (1¼ in) long and, since it runs at a much lower temperature than the exhaust valve guide, it is not completely surrounded by water. All the guides are 14 mm outside diameter, but the inlet and exhaust valve guides are of different lengths. They have a chamfer at their upper ends to form a knife edge to restrict the flow of oil down the valve stems, and a smaller chamfer is machined round their lower ends. Each guide is located by a circlip in a groove a short distance from its upper end, and therefore can be centreless ground. The circlip seats in a small counterbore at the upper end of the housing.

Each valve is undercut 0·4 mm on the diameter between the head and a point 5½ mm inside the lower end of the guide. This is to avoid direct contact between the hottest part of the stem and the oil film in the guide, and also to allow any deposits that may be formed to build up to a thickness great enough for them to flake off easily instead of jamming the valve in its guide. There are three grooves round each stem, just inside the upper end of the guide, to form oil reservoirs. These retain the oil while the engine is stationary and thus reduce guide and stem wear.

Seat inserts, of plain rectangular section, are shrunk into the aluminium alloy head. Since smooth flow is the main requirement in the inlet ports, the internal diameter of the seat inserts is the same as that of the throat of the port. However, the internal diameter of the exhaust valve seats is less than that of the ports; thus, the seats shield the edges of their housings from direct contact with the exhaust gases as they flow out through the ports. This obviates any tendency for the sharp corner round the counterbore, in which the seats are housed, to be burned.

Cylinder head, manifolds and carburettor

The cylinder head casting is of aluminium-copper-silicon alloy. It is of exceptionally sturdy construction because of its compact, rectangular shape: the overall dimensions of the casting are about 10¼ in long by 9⅞ in wide, and 4 in deep between the upper and lower face joints. The combustion chambers are not truly hemispherical, in that they incorporate a shoulder to give a squish effect. This shoulder is on the side of the valves remote from the sparking plug.

Two advantages are obtained with this arrangement. One is a high degree of turbulence which otherwise is almost impossible to obtain with the hemispherical type of combustion chamber unless breathing efficiency is sacrificed. The other advantage is that the plug is nearer to the centroid of the chamber. With a truly hemispherical chamber, the only way to bring the plug as close as possible to the centroid is to place it immediately between the valve seats; this, of course, restricts the size of the valves. The included angle of the valve stem axes in each cylinder is 70 deg and a line bisecting this angle is normal to the plane of the head joint, that is,

VALVE DATA

	Inlet	Exhaust
Material	Heat-resistant, nickel-chrome steel	
Head diameter	30 mm	27·5 mm
Stem diameter	7 mm	
Seat angle	45 deg	
Face width;		
on valve	2·83 mm	2·12 mm
on seat	1·13 mm	1·48 mm
Seat material	Copper aluminium bronze	
Spring length, free;		
inner	31 mm	
outer	33 mm	
Spring length, installed;		
inner	26 mm	
outer	28·2 mm	
Number of coils;		
inner	6·5	
outer	5·5	
Coil diameter;		
inner	16·9 mm	
outer	24·5 mm	
Wire gauge;		
inner	2·5 mm	
outer	3·2 mm	
Valve lift	7·5 mm	
Rocker ratio	1·65:1	
Valve guide material	Grey cast iron	
Valve guide length	47 mm	81 mm
Valve guide inside and outside diameters	7 mm and 14 mm	
Tappet clearance	0·15 mm	0·20 mm
Valve opens	2 deg B.T.D.C.	37 deg B.B.D.C.
Valve closes	40 deg A.B.D.C.	2 deg A.T.D.C.
Ignition timing	Fixed advance 8 deg Automatic advance 28 deg	

it is neither in line with nor parallel to the axis of the cylinder.

As viewed in side elevation, the axes of the sparking plug bosses are vertical but, as viewed in end elevation, they are inclined inwards. They are recessed in the head casting to a depth greater than the length of the plug. The upper ends of the recesses are threaded to receive adaptors, which incorporate spring-loaded plunger type contacts that bear on the ends of the plug terminals. Thus, it has been possible to use standard plugs, despite the depth of the recesses.

The inlet ports of numbers 1 and 2 cylinders and numbers 3 and 4 cylinders are siamesed and pass out to the right of the cylinder head, while the exhaust ports are separate and pass out to the left. This means that the inlet ports of the left-hand bank and the exhaust ports of the right-hand bank are unusually long. However, the arrangement of the exhaust ports is such that they are surrounded by large water jacket spaces, and ducts in the lower face of the head and the upper deck of the cylinder block direct the coolant over the hottest parts. A vertical duct is cored in the front end of the casting to form the oil filler. It has a sleeve pressed into its upper end, and a cap is fitted over the sleeve. Immediately in front of the oil filler is the thermostat housing and water outlet.

A simple cast iron exhaust manifold is bolted to the left-hand side of the block, and the cast aluminium inlet manifold is on the right. The abutting ends of the inlet ports in the head and in the manifold are chamfered to enable a smooth gas flow to be obtained, even if the ports are not exactly in line. A water jacket is incorporated round the manifold and riser. It has two pipe connections: one is from the rear of the head to a banjo union at the base of the manifold, and the other is the outlet back to the pump. The inlet ports are $\frac{31}{32}$ in diameter and their axes are only $1\frac{21}{32}$ in apart, so the inlet manifold is unusually compact. As viewed from above, the main gallery tract is of U-shape, the riser joining it at the centre of the base of the U.

A Solex downdraught carburettor, type 32/30 BI, is fitted. Its jet sizes are as follows: diffuser 20·5, main jet 105, emulsion jet 185, slow running jet 140, and starter device 130. From the $8\frac{1}{3}$ gallon tank, the petrol is lifted by a mechanical pump to the carburettor.

Water pump and cooling system

The fan and water pump are driven at 1·36 times engine speed. To ensure that water cannot leak into the bearings, an unusually complex pump layout has been adopted. An aluminium alloy casting houses the $2\frac{5}{32}$ in diameter brass rotor, which is pinned on to a $\frac{3}{8}$ in diameter spindle, and the rear end of the housing is closed by a $\frac{7}{32}$ in thick steel plate. The whole assembly is bolted to the front end of the cylinder block casting.

Two white metal lined bronze bushes in the nose of the body casting carry the spindle, the $\frac{3}{32}$ in long annular space between them forming a water trap. The rear bush is $\frac{13}{32}$ in long, while the length of the front one is $1\frac{29}{32}$ in. A spring-loaded rubber water seal is interposed between the rotor and the flanged end of the rear bush. If any water should pass the seal and seep through this bush, it is trapped in the annular space and then drained away through a vertical hole drilled from the lower face of the casting.

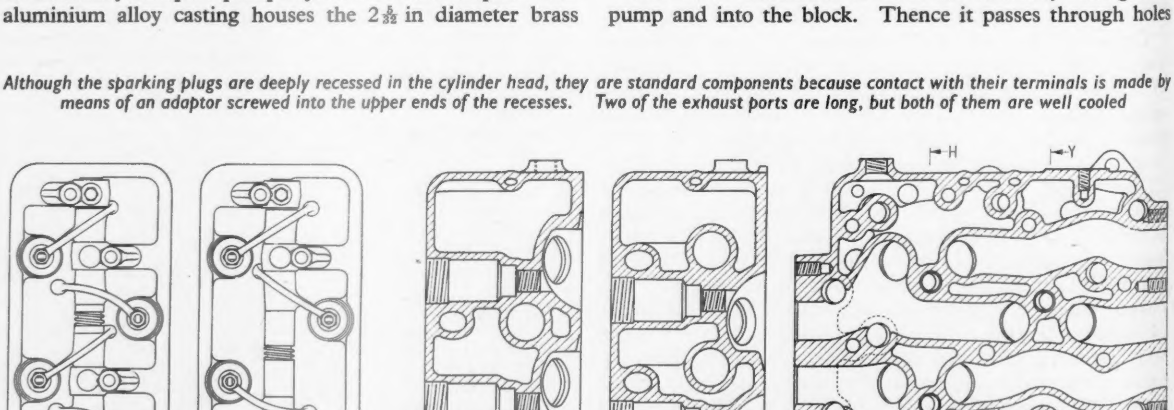
Two ball bearings, with their centres approximately $\frac{7}{8}$ in apart, are assembled on to the nose of the casting. These bearings carry the pulley and fan. A distance piece separates the inner races of these ball bearings and they are retained by a ring nut screwed on to the front end of the nose of the body casting. The outer race of the rear bearing is housed in a counterbore in the pulley, while that of the front bearing is housed in the boss of the cast aluminium fan. The fan is secured to the front face of the pulley boss by three countersunk set screws. Its concentricity, relative to the axis of the spindle, is assured by the close tolerances on the periphery of the nose-piece.

Axial location of the whole assembly is effected by the outer race of the front ball bearing, which is clamped between a shoulder in its housing in the fan boss and a steel ring in a counterbore in the forward end of the rear ball bearing housing. The drive to the pump spindle is effected through a steel insert cast in the centre of the fan boss: diametrically opposed flats on the end of the spindle register between flats in a hole through the centre of the steel insert. The extreme front end of the spindle is threaded to receive a nut that is used, in conjunction with a distance tube, to retain the whole assembly.

Lubrication is effected through a grease nipple on top of the housing—it is of interest to note that this and the front wheel hubs are the only gun-lubricated points on the whole of the engine and chassis. The grease passes down a drilled hole into an annular space in the housing for the front white metal lined bush. From this space it goes in two directions: some passes radially inwards through holes to the bearing surface round the spindle, and the remainder goes outwards through holes to the space between two ball races.

The water circulation is from the radiator, through the pump and into the block. Thence it passes through holes

Although the sparking plugs are deeply recessed in the cylinder head, they are standard components because contact with their terminals is made by means of an adaptor screwed into the upper ends of the recesses. Two of the exhaust ports are long, but both of them are well cooled



in the top deck into the head. These holes are positioned so as to direct the coolant to the hottest parts of the head, that is, round the exhaust ports and sparking plug bosses. The main return is past the thermostat valve in the outlet at the front end of the head, but there is a subsidiary outlet through the rear of the head, through the water jacket round the induction manifold and thence through another pipe to the pump.

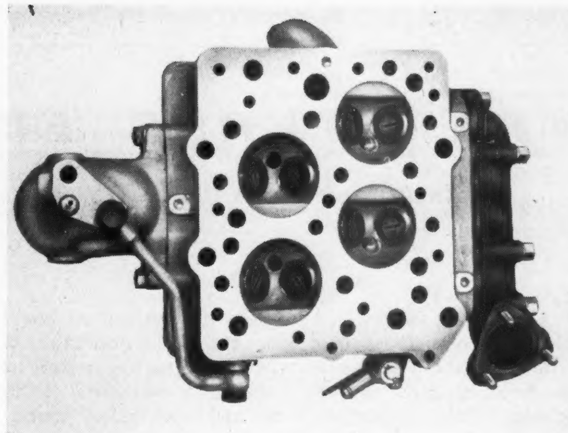
Lubrication system

Whereas in the Ardea engine, the oil pump was mounted with its axis horizontal, in the cylinder block casting and was, therefore, high above the oil level in the sump, this engine has its oil pump partly submerged below the oil level. The latest arrangement is better for a number of reasons. One is that the pump is self-priming, so that there is no danger of oil starvation if the engine has been stationary for a long period. Another is that when the oil pump is mounted high up in the crankcase, the discharge from the relief valve in the pump body tends to be churned up by the rotating crankshaft. This may cause excessive aeration and even loss of power due to oil drag. In the Appia engine, a spring-loaded ball type relief valve by-passes the oil into the inlet side of the pump. This lightens the load on the pump and greatly reduces the possibility of cavitation in the suction pipe.

A spiral gear on the camshaft drives the pump and distributor in the usual manner. The driven gear is of hardened nickel-chromium steel (15 NC5 Cmt) and is pinned to the spindle. At the upper end of the spindle a sleeve coupling forms the connection to the contact breaker and distributor unit. A hardened steel thrust washer is interposed between the lower end of the driven gear and a flanged phosphor bronze bush. This bush, which is $\frac{3}{8}$ in inside diameter by $1\frac{5}{16}$ in long, is inserted from above into its housing in the crankcase. A hole is drilled in it to receive the end of a countersunk set screw assembled radially into the housing to lock it in position. Immediately below this bush a $\frac{3}{8}$ in long sleeve is pinned to the spindle. The function of this sleeve is to act as a stop to prevent the spindle from being forced upwards by oil pressure. At the lower end, the spindle projects into the pump housing and is tongued to register in a slot in the upper end of the short driving spindle of the pump. Both pump gears are $\frac{5}{8}$ in long by $1\frac{1}{16}$ in diameter; the driving gear is keyed on to its spindle, while the driven gear is pressed on.

Circulation

The large diameter, inverted-dish type pick-up is formed by an aluminium alloy casting and incorporates a strainer gauze. It is mounted with its lower face only about $\frac{5}{32}$ in above the base of the sump and is bolted directly to the pump. If an external filter is fitted, the lubricant passes through it before going into the 10 mm diameter gallery in the right-hand side of the crankcase, and the connection to the oil pressure gauge is taken from the top of the filter. On the other hand, if there is not a filter, the oil passes directly into the gallery. With this arrangement, the oil pressure gauge connection is fitted in the right-hand end of a 10 mm diameter passage drilled transversely in the front wall of the crankcase, to form a T-junction with the gallery. The left-hand end of this transverse passage is under the front main journal bearing, to which it is connected, and another vertical duct is taken from it to a relief valve above the timing drive. From the relief valve, which lifts at 50 lb/in², the discharge falls on to the timing drive chain. The function of this valve, of course, is to restrict the flow to the timing chain so that the main bearings are not starved of oil when the pressure is low immediately after the engine is started, particularly after long periods of standing.



The incorporation of a shoulder in each hemispherical combustion chamber not only gives a satisfactory degree of turbulence, owing to the squish effect, but also enables the plug to be positioned more readily near to the centroid of the chamber: both features improve combustion

Details of the feed to the front main journal are as follows. From the left-hand end of the transverse passage, the oil passes into a space between the crankcase wall and the flange of the main journal bearing housing. Thence it passes through a vertical hole in the housing, which is in line with the radial hole in the bearing. The oil then passes into a groove that extends about three-quarters of the way round the bearing surface. At each end of this annular groove is a longitudinal spreader groove. From the annular groove, the oil is fed into the hollow crank pin by four radial drillings. The two adjacent big end bearings are served from this journal by holes drilled through the shaft. A $2\frac{1}{2}$ mm diameter hole from one of the longitudinal spreader grooves feeds oil up to the hydraulic chain-tensioner and to a groove round the end of the spindle that carries the eccentric jockey sprocket. Radial holes take the lubricant into an axial duct in the spindle, whence it passes out again through more radial holes to the bearing surfaces.

The rear main journal is served by a pipe between the lower end of a vertical passage drilled up to the main gallery and a banjo connection bolted up to the base of the main journal bearing housing. Lubricant goes from this pipe connection, up through holes to a groove round the bearing surface, and thence into the hollow journal. Drillings in the shaft take the lubricant to the two adjacent big ends. The big ends are lubricated by splash in the usual manner.

There are four radial holes in the rear journal, and they intermittently come into line with a 5 mm diameter vertical duct serving the camshafts and overhead rocker shafts. The camshafts are fed through an 8 mm diameter passage drilled horizontally in the rear wall of the crankcase. From the ends of this passage, the oil passes into the rear camshaft bearing, and thence along the hollow shaft and through radial drillings to the other two bearings.

At the joint face between the crankcase and cylinder block, a tubular dowel is fitted in the vertical oil passage to the rocker shafts. The hole through the centre of this dowel is $2\frac{1}{2}$ mm diameter, so it restricts the flow to the rocker shafts. The dowel also serves to locate the crankcase relative to the cylinder block and helps to seal the junction between the oil ducts in these two components. In the cylinder head, the passage is Y-shaped, the two branches going to a rocker pedestal on each side and thence into the hollow shafts. Radial holes in the shafts serve the bearing surfaces in the rockers. Oil returns through the holes for the push rods.

TRANSVERSE FORCES ON TYRES

Analysis of Results Obtained From Road Tests With Several Different Vehicles

Dr. A. Chiesa*

Four main factors lead to transverse loading on tyres. These are centrifugal force when the car is negotiating a bend, inclination, or camber, of the road surface normal to the direction of motion, side winds, and oscillations of the vehicle. These transverse forces are balanced, of course, by others due to deflection of the front wheels at a certain angle with respect to the direction of motion. This angle is known as the slip angle and its value influences to a considerable degree the rate of tyre wear.

The determination of the transverse forces acting on a car in motion is difficult because of the large number of variables involved, and the rapidity with which some of them change. In the investigation described in this article, some general experimental conditions, such as car type, driver, road, and average speed, were selected and kept constant, so that the rapidly varying nature of the transverse forces could be more closely examined. This made possible the development of methods of measurement and analysis which, even if not of extreme accuracy, nevertheless gave comprehensive data in a compact form that will be of use to designers.

Inertia type accelerometers placed transversely in the car were employed to take measurements from which the acceleration forces could be calculated. The order of magnitude of the errors introduced by this method is discussed later, but seems acceptable for this purpose. Signals from the accelerometers were fed to recorders, or to special electronic integrating devices.

Apparatus and method of measurement

The inertia accelerometers used were of the strain gauge type with a linear response up to 100 c/sec and an adequately unidirectional sensitivity. They were placed transversely on the floor of the car and used as follows:

(a) For direct recording of the acceleration curve, by means of an amplifying recorder. A graphical analysis of the recordings was then made.

(b) In conjunction with an electronic integrator and a stop watch, for determining directly the average value of the transverse acceleration throughout a whole run.

Direct recording method. This method shows the way in which the acceleration, and hence the transverse force, varies with time as the car moves along the test route. Peak forces can be positively traced to incidents occurring during the test run. Data can be obtained concerning the ratio of the time during which the acceleration exceeds a given value to the total test time, and also the number of times this value is exceeded per unit length of test run.

There are, however, several disadvantages to this method. Among these is the fact that the apparatus, which is very heavy, must be carried in the car. To obtain sufficiently reliable and exhaustive results, that is, corresponding to a sufficiently long test run, it is necessary to examine a great length of record. For example, with the ink recorder used, a test run of 10 km at an average speed of 72 km/hr required 10 m of recorder paper. The analysis of the results is therefore long and tedious. Finally, the length of the test run is limited, by the capacity of the batteries, to one or two hours.

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Integration method. The electronic integrator, Fig. 1, is of a type previously described.[1] It records, in the conventional way, the average of the absolute values of electric signals, of any form, fed to it from a transducer, such as an accelerometer, piezometer, dynamometer, or microphone. The electric signal obtained from the transducer is first amplified and then, if the signal is an alternating one, it is rectified; finally it is integrated. By dividing the final value of the integral by the time over which the measurements were taken, the average value of the signal can be obtained.

Advantages of the instrument are that it is light, vibration-resistant and, being battery fed, can be used in any car without special auxiliary equipment. When used with normal type batteries, it can be run for several hours at a time. Therefore, continuous measurements can be made on long test runs, even up to several hundred kilometres. The results are continuously recorded and elaborate statistical or graphical processing is not required. However, the main disadvantage of this equipment is that it does not give the peak values of accelerations.

Calibrating and checking. To ensure that precise measurements were obtained, the apparatus was calibrated at the beginning and end of each test run. Checks were made on measurements obtained both with the recorder and with the integrator. This was done simply by checking the reading against the acceleration of gravity, measured by turning the accelerometer into the vertical plane. Only very slight variations were observed throughout all the tests; nevertheless, the individual calibration values were taken into account in the calculations.

It might be thought that errors could still arise from the fact that calibration was normally carried out with the car at a standstill, whereas when the car is in motion, its unsteadiness, particularly the high frequency oscillations, might interfere in various ways with the correct operation of the apparatus. Trouble might be caused, for example, by microphonic effect of the tubes, or defective contacts. However, errors of this nature were avoided by designing the apparatus

Fig. 1. The electronic integrator that was used for the tests

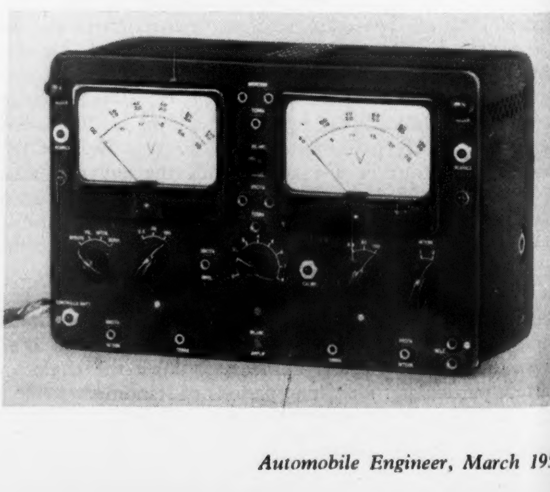

TABLE I

Vehicle	Test route	Length km	Time min	Time sec	Average speed km/hr	Average transverse accelera-tion m/sec²	Average transverse force per tyre kg	Accelera-tion limits m/sec²	Corres-ponding transverse force limits per tyre kgp	Ratio of time in which limit is exceeded to total time per cent	Number of times limit is exceeded per unit length km
Car 1	**Lakeside route**										
	Lecco-Colico	45	51	30	52·4	0·70	29	2	82	7·67	2·09
								3	124	1·18	0·51
								4	166	0·03	0·04
	Colico-Menaggio	20	23		52·1	0·66	28	2	82	6·30	2·10
								3	124	1·23	0·52
								4	166	0·25	0·16
								5	217	0·04	0·05
Car 2	**Lakeside route**										
	Lecco-Colico	45·7	45	13	60·6	0·98	43	2	88	9·35	2·95
								3	132	4·10	1·34
								4	176	0·61	0·37
								5	220	0·02	0·02
	Colico-Menaggio	29·8	35	33	50·3	0·82	36	2	88	7·42	3·08
								3	132	2·20	1·28
								4	176	0·52	0·41
								5	220	0·28	0·20
								6	264	0·05	0·03

TABLE II

Vehicle and driver	Test route	Length km	Time hr	Time min	Time sec	Average speed km/hr	Average transverse acceleration m/sec²	Average transverse force per tyre kgp
	Lakeside route							
Car 1	Como-Colico	60	1	11	30	50·5	0·92	36
	Colico-Como	61·4	1	14	3	49	0·75	29
	Colico-Lecco	40		37	30	64	0·96	38
	Lecco-Colico	45·3		44	5	61·7	0·98	38
	Autostrada							
Car 1	Milano-Torino	128·3	1	40	5	76·9	0·31	12
	Torino-Milano	129·8	1	44	35	74·4	0·33	13
	Lakeside route							
Car 2	Como-Colico	60·8	1	3	13	57·6	1·24	49
	Colico-Como	61	1	1		60	1·39	55
Driver A	Colico-Lecco	45		37		73	1·63	65
	Lecco-Colico	45·2		37	55	75·5	1·63	65
	Autostrada							
Car 2	Milano-Torino	130	1	7		116·4	1·21	48
Driver A	Torino-Milano	127	1	2		122·7	1·18	47
	Lakeside route							
Car 2	Como-Colico	62	1	11	41	51·9	1·03	41
	Colico-Como	61·6	1	11	9	52·1	0·96	38
Driver B	Colico-Lecco	45·1		42	40	63·4	1·26	50
	Lecco-Colico	45·3		42	7	64·5	1·31	52
	Autostrada							
Car 2	Milano-Torino	130·1	1	9	30	112·3	1·08	43
Driver B	Torino-Milano	129·8	1	8	45	113·3	1·18	47
	Autostrada							
Truck	Milano-Torino	123·8	2	11	57	56·4	0·96	83
	Torino-Milano	123·6	2	9	23	57·3	1·19	103
	Road Circuit							
	Milano-Brescia		4	18	20	58·1	0·81	70
	Piacenza-Milano	250						

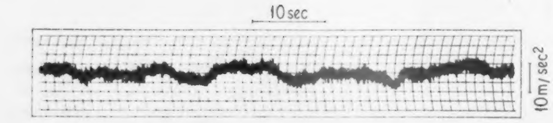

Fig. 2. A section of a record of the transverse accelerations

in such a way as to render it practically insensitive to vibration. Also, the calibration was repeated with the accelerometer turned into the vertical plane and with the car travelling at 130 km/hr along a straight flat road, with a fairly rough surface; the results were in agreement with those obtained for static calibration.

Tests and results

Conditions selected for the measurements. Of the many factors that affect the transverse forces, the following were taken into consideration: type of vehicle, that is, fast passenger car, or transport vehicle; driving habit, fast or moderate speed; type of road, mainly straight or winding. To gather broadly orientative data, the programme of tests decided on included diverse combinations of these test conditions, as follow:

(a) Type of vehicle:

Car 1	Normal touring type; engine swept volume 1,400 cm^3; loaded with four passengers and 50 kg (integrator) or 110 kg (recorder) apparatus
Car 2	Fast touring type; engine swept volume 1,900 cm^3; loaded as above
Truck	Medium speed type; fully loaded, carrying 2,500 kg

(b) Driving habit:

Driver A	Calm and safe
Driver B	Fast and nervous

(c) Type of test road:

Autostrada	An almost straight run on the autostrada between Milano and Torino; distance 130 km
Lakeside run	A very winding route on the lake-shore road between Como, Colico and Lecco; distance 106 km
Road circuit	Normal level road on the circuit Milano, Brescia, Piacenza, Milano; distance 250 km

Summary of the results. Since the peak values recorded did not exceed 5 m/sec^2, except for a single case in which 6 m/sec^2 was reached, five limit values (2, 3, 4, 5 and 6 m/sec^2) of transverse acceleration were chosen for the analysis by the direct recording method. Some of the results from the recordings are shown in Table I. A particularly unsteady section of the acceleration curve is reproduced in Fig. 2. The results obtained with the electronic integrator are all average values, which are shown in Table II.

Comments on results

Deviation from the average value. Table I shows that the acceleration does not deviate much from the average value, even in tests that were carried out on the most tortuous route. For example, the total time during which a transverse acceleration of three times the mean acceleration occurred was far less than eight-hundredths of the duration of the run. Therefore, it may be concluded that adequate information can be obtained from direct measurements of the average values, obtained with the electronic integrator.

Average values. The average transverse forces, as determined with the electronic integrator, are of the following orders of magnitude:

Car 1	10-40 kg per tyre
Car 2, driver A ..	45-65 kg per tyre
Car 2, driver B ..	35-55 kg per tyre
Truck	70-100 kg per tyre

These values expressed as percentages of the total load carried by each tyre for the three vehicles are as follows:

Car 1	2-10 per cent
Car 2, driver A ..	11-17 per cent
Car 2, driver B ..	9-14 per cent
Truck	8-16 per cent

Influence of the type of run and speed. Although the various test runs were chosen with widely different characteristics, that is, one mainly straight, one normal and the other with many curves, the transverse loads obtained do not differ much from each other. This is especially noticeable for Car 2 and for the truck—with Car 1, the test run on the autostrada was performed at a moderate speed. The fact that the transverse accelerations do not differ greatly, in spite of the difference in the characteristics of the test runs, can be explained as follows. High speeds can be maintained on mainly straight roads and this causes a considerable increase in the transverse acceleration every time another vehicle is overtaken and for every slight change of direction, whereas on winding roads the speed that can be maintained in a curve is necessarily correspondingly lower.

Influence of the driver. A different average value of the transverse accelerations was obtained with each of the two different drivers on Car 2. On the other hand, each driver's average was always in the same order of magnitude for all the different types of test run. From this it can be concluded that each driver controls the car more or less subconsciously in such a manner as to maintain a constant average value of transverse acceleration on any type of road. This value is probably determined by a combination of physiological and psychological factors, such as comfort, ability and sense of responsibility of the driver, and familiarity with the car.

Theory of the measurement of the transverse forces

Determination and causes of transverse forces. As previously noted, when a car is in motion on a road, the causes of the transverse forces on the tyres can be summarized as follows:

(a) Transverse component, of the road reaction to the weight of the car, due to transverse slope or camber of the road surface
(b) Centrifugal forces arising when the vehicle follows a curved path
(c) Transverse components of general oscillations of the car body
(d) Action of side winds

To determine experimentally the contribution of the first three items to the overall transverse force, the transverse acceleration was simply multiplied by the mass of vehicle. This does not take into account the contribution of side winds, the influence of which is generally limited and will be considered briefly later. The rate of wear was of the same order for each of the tyres and this was taken to indicate that the forces also could be considered to be equally divided between the four. [Presumably the author is referring to the average force distribution over a relatively long period of time—Ed.]

Inaccuracies in the measurement of the transverse acceleration. For the measurement of the transverse acceleration, an accelerometer was mounted on the floor of the car in such a way that, when the vehicle was stationary on a horizontal plane, the axis of maximum sensitivity of the accelerometer was horizontal, and normal to the longitudinal axis of the car. This arrangement is much the simplest of the many that could be used; however, as the car body tilts when subjected to the forces to be measured, or for other reasons, it gives rise to inaccuracies, which are discussed below.

Transverse slope of the road. If the car travels along a straight road having a transverse slope, the average value of the lateral force on the tyres is given by $mg \sin \alpha$, where m is the mass of the car, g acceleration due to gravity and α the angle of tilt assumed by the car. Under these conditions, the accelerometer gives an indication of $g \sin \alpha$, and consequently shows a true measure of the average lateral force. An idea of the magnitude of the forces arising from this cause can be obtained from the fact that a transverse inclination of 1 deg, of a car weighing 1,500 kg, produces a transverse force on each tyre of $6{\cdot}25\ kg_p$. Surveys carried out on many types of road have shown that slopes of this magnitude are fairly common; slopes of 2-3 deg are rarer, while transverse slopes of 4-5 deg are exceptional.

Swaying of the car while negotiating curves. When the car sways, owing to the effect of centrifugal force, an accelerometer rigidly attached to the body does not give a reading from which the transverse force can be obtained directly. The curve may be flat or banked or may even have two slopes, one transverse, due to the banking, and the other tangential to the curve, which, of course, has no effect on lateral forces. A transverse slope affects the accelerometer in the manner already discussed. As a first approximation, this can be considered as not interfering with the effect of centrifugal acceleration, which therefore may be studied as if there were no transverse slope.

If a is the centrifugal acceleration in the horizontal plane at the centre of the car, the overall transverse force is given by ma, where m is the mass of the car. If α is the angle of transverse tilt of the car body, due to sway caused by the centrifugal force, the reading δ of the accelerometer will be:

$$\delta = a \cos \alpha + g \sin \alpha$$

Thus the reading will be high. Since the tilt angle α, which causes this error, depends on the value of a, it is possible to calculate a correction coefficient, which, of course, will be less than unity. In the following calculations, the suffix p indicates the condition on the axle and the suffix $_T$ indicates that on the body, Fig. 4, and the following symbols are used:

α_p = angle of tilt of axle
α_T = angle of tilt of body
x_p = axle sway, that is, difference between the heights of the ends of the axle from the horizontal plane of the road
x_T = body sway
l = track of vehicle
h = height of centre of gravity of car
k_p = stiffness, or rate, of the two tyres acting together on one side of the car
k_S = stiffness of the two suspension springs acting together on one side of the car

Fig. 4. *Diagram showing the nature of the relative movements between the body, axle and road, when the vehicle is negotiating a curve*

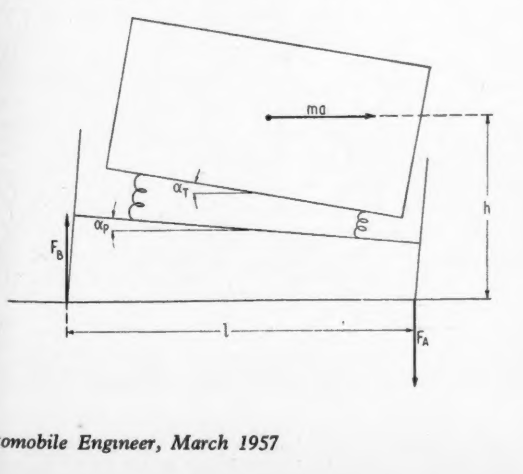

Fig. 3. *One of the vehicles negotiating a curve during the tests*

k_T = total stiffness on one side of the car
N/m = Newtons per meter of stiffness ($1N \approx 0{\cdot}1\ kg_p$); this is the metric unit of stiffness

The tilt angles are:

$$\alpha_p = \tan^{-1} \frac{x_p}{l}$$

$$\alpha_T = \tan^{-1} \frac{x_T}{l}$$

The accelerometer reading is then:

$$\delta = a \cos \left(\tan^{-1} \frac{x_T}{l}\right) + g \sin \left(\tan^{-1} \frac{x_T}{l}\right)$$

Assuming the cosine of the small angle to be unity, and $\sin \alpha = \tan \alpha = \alpha$ radians:

$$\delta = a + g\frac{x_T}{l}$$

$$= a + g\frac{2mh}{l^2 k_T} a$$

$$= a\left(1 + g\frac{2mh}{l^2 k_T}\right)$$

$$= a\frac{1}{\eta_T}$$

Hence:

$$\eta_T = \frac{1}{1 + \frac{2mhg}{l^2 k_T}}$$

where η_T is the required correction coefficient for an accelerometer mounted on the body.

A similar analysis, applied in the case where the accelerometer is mounted on the axle, gives:

$$\eta_p = \frac{1}{1 + \frac{2mhg}{l^2 k_p}}$$

Since $k_T < k_p$, this correction factor is smaller than that previously obtained.

Application of the above formulae to one of the cars used for the tests gives the tilt, in metres, resulting from a transverse acceleration of value a, in m/sec². In this car: m = 1,500 kg, h = 0·8 m, l = 1·4 m, k_S = 55,000 N/m, k_p = 300.000 N/m, and k_T = 46,000 N/m.

For the axle: $x_p = 0{\cdot}0057a$
For the car body: $x_T = 0{\cdot}0311a$

Consequently, the tilt angles, in radians, are respectively:

$\alpha_p = 0{\cdot}0041a$
and $\alpha_T = 0{\cdot}0222a$

and the correction coefficients are:

$\eta_p = 0{\cdot}96$
and $\eta_T = 0{\cdot}80$

That is, the error introduced in the accelerometer reading is an increase of 4 per cent if the instrument is mounted on the axle, and an increase of 26 per cent if it is mounted on the car body.

To check these formulae, an experiment was carried out to determine by photography the angle of tilt of a car negotiating a curve, of 10·5 m radius at a speed of 6·9 m/sec, 24·7 km/hr. This angle was found to be about 6 deg, Fig. 3. The transverse acceleration is $a=4{\cdot}6$ m/sec^2 and, from the formulae, the angle of tilt $\alpha_T=0{\cdot}022a=0{\cdot}022\times4{\cdot}6=0{\cdot}102$ *radn*$=5$ deg 50 min, which agrees well with the value obtained experimentally. Also, the tilt angle of the axles, according to the formulae, is $\alpha_p=0{\cdot}00407a=0{\cdot}00407\times4{\cdot}6$ $=0{\cdot}01875$ *radn*$=1$ deg 5 min.

Summarizing, it can be said that the error introduced by the swaying of the car when negotiating a curve on a flat road-surface is always an increase, since $\eta<1$. It is small if the accelerometer is fixed to the axle, but is fairly large if the instrument is fixed to the body of the car.

Transverse oscillations of the car. During its motion, the car oscillates vertically and transversely due to irregularities of the road and of driving. The rear axle of the car oscillates at a frequency of 10-12 c/sec in a vertical plane, and also at a high frequency, as much as 100-200 c/sec, owing to the motion of the gears in the differential and to elastic deformation of its constituent parts. At the high frequencies, the amplitude of the accelerations may be very high, even though the amplitude of motion is small.

On the other hand, the body of the car oscillates at lower frequencies, with accelerations of orders of magnitude about one tenth of those of the axle. Experience of tyre wear, which is known to be much increased on winding routes, suggests that transverse oscillations must play a secondary role so far as wear is concerned. Consequently, they need not be taken into account in this investigation. Isolation of the accelerometer from these oscillations is effected, at least partly, if it is mounted on the structure inside the car body.

Because of the transverse oscillations, it seems impossible to make a quantitative analysis of the error introduced in the measurement of centrifugal force. This is certainly the weakest feature of the analysis and measurement procedure.

Side winds. The effect of side wind on a car could be measured by experimental methods. However, in view of the order of magnitude of the wind pressure on the body under normal conditions its contribution to the transverse forces is always relatively small by comparison with that due to the other factors. Therefore, a calculated figure can be used without unduly affecting the accuracy of the overall result.

Except under exceptional conditions, however, the average speed of the component of the wind that strikes the car broadside on rarely exceeds 3-4 m/sec. This speed is appreciable, and the resultant pressure on the body is about 5-7 kg$_p$/m^2; nevertheless, the overall transverse force, as will be shown later, is from 2-10 times greater than that due to side winds.

Even if it is desired to determine the transverse force acting on the tyres in very windy conditions, it could probably be obtained with sufficient accuracy by calculation. Consider, for example, a car travelling over a certain route where the transverse component of the wind is 8-10 m/sec, which corresponds to a very strong side wind. The pressure exerted by this wind on the body is about 16-20 kg$_p$/m^2. This gives rise to a transverse force of about the same order as those due to the other factors and is therefore no longer neglible. [Useful information might be brought to light by further investigation on the overall aerodynamic effect as opposed to approaching the problem simply with regard to the lateral component of the wind velocity—Ed.]

The effect of the wind on the total transverse force is particularly noticeable in gusty conditions. In spite of the corrective action taken by the driver, the car may even skid sideways. Measurements under these conditions were carried out by Lay and Left[2], who observed that when a car in motion is subjected to sudden gusts of wind, the corrective action of the driver may give rise to average transverse forces of the order of 20-25 kg$_p$ per tyre.

Another point to be considered is the effect of the wind on the measurement of the transverse forces due to other causes. The wind causes an inclination of the car body, and this is superimposed on the sway arising from the other causes previously discussed. This introduces in the accelerometer reading an error, the magnitude of which is best illustrated by a practical example.

If a car in motion is subjected to a transverse wind having an average velocity of 4 m/sec, the overall transverse force acting on the car is about 30 kg$_p$. From the equations and data previously given for one of the test cars, the angle of tilt can be shown to be about 19 sec. The corresponding error in the accelerometer reading, multiplied by the mass of the car acting on each tyre, is about 2·3 kg$_p$; this can be neglected because the values of the transverse force due to other causes are about 5-30 times greater.

Conclusions

By using the experimental method described, the mean values of the transverse forces acting on a moving vehicle can be evaluated. Although the results obtained are only approximate, they are nevertheless adequate for the purpose of estimating loads, stresses, and probably rates of wear of vehicle components. Factors such as road type, car type and driving conditions are important; and the direct method of measurement is justified, notwithstanding the difficulties discussed. The author wishes to thank Prof. Valentino Zerbini for the valuable advice given him in the preparation of this paper, and Pirelli Societá per Azioni for permission to publish the work.

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PHOTO-RECORDING PAPER

A NEW ultra-fast, high contrast photo-recording paper, termed Lino-Writ 4, has recently been introduced by E. I. du Pont de Nemours and Co. Inc., of Wilmington 98, Delaware, United States of America. It records oscillographic test data at frequencies and recording speeds higher than hitherto possible. Lino-Writ 4 is the newest in the series of orthochromatic photo-recording papers that this manufacturer has developed.

Each type of paper in the series is adapted to specific conditions of use. The type that has just been introduced is approximately 25 per cent more sensitive than Lino-Writ 3. Not only does this permit the recording of test data at ultra-high frequencies, but also the paper can be run through the machine at a high speed to stretch out the oscillograph record or trace, with consequent improvement in clarity and ease of analysis. Also, the new paper is only one-quarter of the thickness of the others: this means that greater lengths can be carried in rolls accommodated in the oscillograph. Either conventional or rapid stabilization processing methods can be used for all four of the Lino-Writ series. The papers are produced in standard sizes to fit the majority of the popular oscillograph machines.

EQUIPMENT FOR AUTOMATION

Sequence Control Units and Ancillary Equipment, Giving Flexibility and Rapid Re-Setting of Automated Tooling for Batch Production

By the automation of a production line the specific rate of production is increased as a consequence of the reduction in handling and transfer times between machines and the increase in the operating speed of the machines made possible by the faster flow of the work. Only in instances where production requirements are on a scale to ensure virtually continuous operation at the enhanced rate, however, will the heavy capital expenditure on special-purpose machine tools and transfer equipment be economically justifiable, or even tolerable. If sufficient production is attained in a percentage of the hitherto normal working time, the rate of machine utilization may well be lowered and thus the line will become relatively inefficient.

The pattern of automation introduced into American industrial plants for exceptionally long production runs would appear to be applicable only in comparatively few instances in British industry, which is characterized by shorter production runs and demands greater operational flexibility. Less highly specialized machines, facilities for the rapid change-over of tooling, quick re-setting of operating sequences, and adaptable handling, transferring and conveying equipment are essential if the advantages of automation are to be realized on short-run or batch production. Versatile pneumatically or hydraulically operated equipments are available which make it possible to design transfer mechanisms, automatic feeders, loaders, unloaders, handling devices, and controls that are relatively flexible in operation. The conventional method of controlling pneumatically or hydraulically actuated operations has been by the use of solenoid valves and cam-operated switches. These were liable to require considerable maintenance and necessitate skilled labour and an undesirably long period of time for re-setting when changing over the line or the machine to run on a different component.

To meet these requirements, the Automotion Division of Industrial Guarding Equipment Ltd., Court Road, Birmingham, 12, has designed and developed a variety of devices for the control and the protection of machines and tools. They are now available as standard units that can be embodied in schemes for the automation of production. It may be noted, in passing, that the firm prefers the term "automotion" to "automation," as being more accurately descriptive of the automatic movement of components to suit a series of operations.

An outstanding development is the "Masterotor" sequence control unit, Fig. 1, which was initially designed for the control of automated tooling in power presses to give complete flexibility of operation where short tooling runs involve frequent re-setting. Its advantages have since been realized for the control of machine tools and the timing of sequential processing or treatment programmes. A standard unit can be used with any machine in which the movement of

Fig. 1. The Masterotor automatic sequence controller comprises an enclosed switchboard and a rotary switch unit

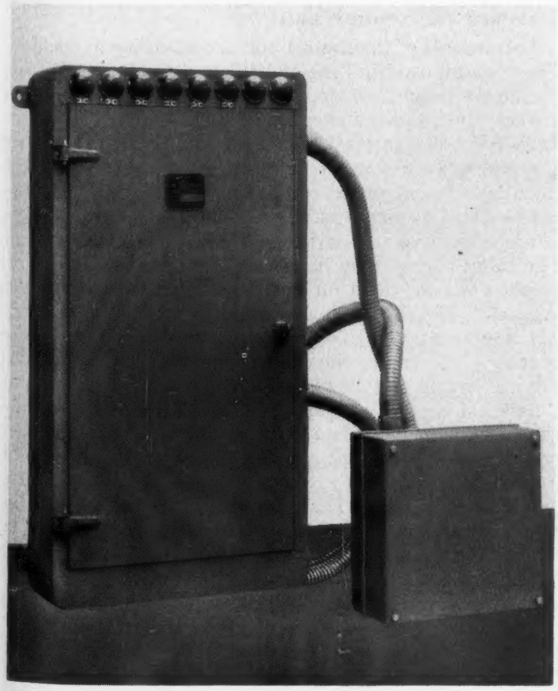
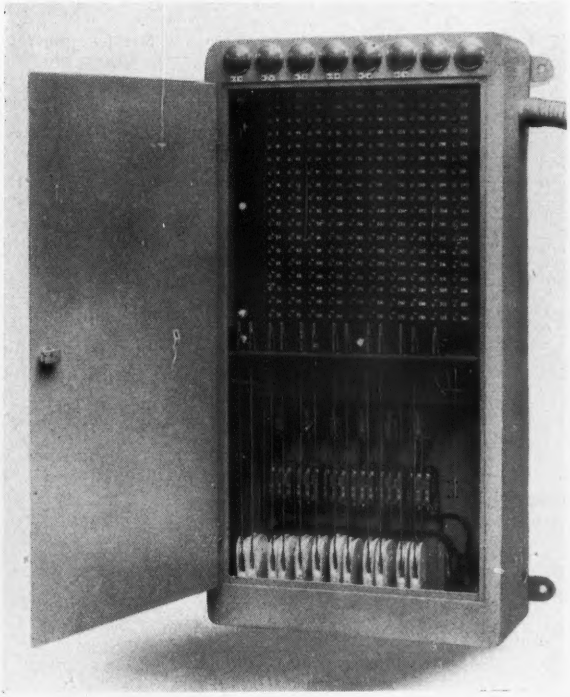


Fig. 2. Control panel opened to show the 180 sockets and the counter-weighted jack plugs



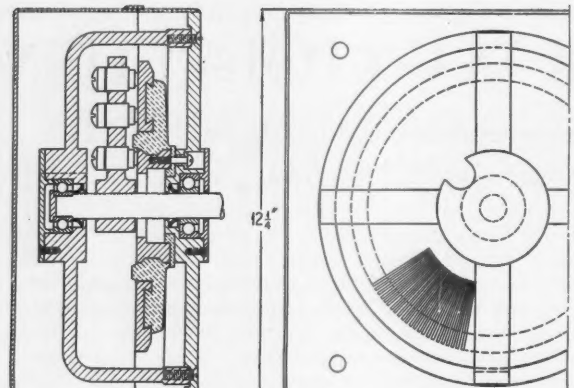
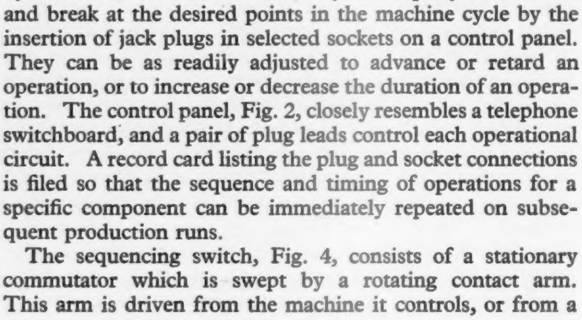


Fig. 3. Arrangement of sequence switch. On the rotating arm two carbon brushes sweep the commutator segments and one brush the annular central contact

the work through its various operations, and the control of the machine, is initiated by a series of electrical circuits, making and breaking at predetermined points in the machine cycle. These circuits can be easily and rapidly set to make and break at the desired points in the machine cycle by the insertion of jack plugs in selected sockets on a control panel. They can be as readily adjusted to advance or retard an operation, or to increase or decrease the duration of an operation. The control panel, Fig. 2, closely resembles a telephone switchboard, and a pair of plug leads control each operational circuit. A record card listing the plug and socket connections is filed so that the sequence and timing of operations for a specific component can be immediately repeated on subsequent production runs.

The sequencing switch, Fig. 4, consists of a stationary commutator which is swept by a rotating contact arm. This arm is driven from the machine it controls, or from a

Fig. 4. The sequence switch with cover removed to expose the 180-segment stationary commutator and the rotating contact arm



synchronous motor controlled by the machine, to make one complete revolution for every complete cycle of the machine. Segments of the commutator are each connected to a socket on the control panel, numbered in degrees to correspond with the appropriate angular position of the segment. By plugging into a socket, say, number 20, an electrical current will flow momentarily through a circuit as the rotating contact arm passes over segment number 20; that is, as the machine completes 20 degrees of its cycle.

It is necessary that this current be held on to initiate some work by electrical, pneumatic, or hydraulic equipment for a specific period of the machine cycle. To this end the sequencing switch is arranged to control a series of monitors, one being provided for each circuit. These monitors each comprise one relay with a normally closed contact and another with a normally open contact; wired as shown in Fig. 5. From this it will be seen that if plug A of the monitor (a red plug in the circuit diagram, Fig. 6) is inserted in any numbered socket, the circuit will be completed at that particular point in the machine cycle to energize the relay coil R1A and close contacts A1. The external circuit to initiate any desired operation is made by a further set of contacts on relay R1A. This enables a high-voltage circuit to be used for operating purposes whilst retaining a low-voltage for actuation of the control unit.

By inserting plug B of the monitor (a black plug in the circuit diagram, Fig. 6) in a numbered socket, the circuit will be at that position completed to energize relay coil R1B. This will open contacts B1, breaking the circuit through relay R1A and, consequently, the external operating circuit. Thus plugs A and B determine respectively where the operating circuit is closed and opened relative to the machine cycle. With a number of monitors a similar number of operating circuits can be controlled to make and break at desired points in the cycle and in correct sequence. Adjustment of the timing and the period of the various operations is effected by merely moving the jack plugs to different sockets. This enables one Masterotor unit to be readily adaptable to control different operations and tooling on a given machine and, if necessary, to be transferred to control a different machine.

Standard Masterotor units

Two models of the control unit are produced as standard. The 8-circuit model, Type 180/8, has a panel of 180 sockets numbered from 2 to 360 and a sequencing switch with 180 segments, giving a selection of "On" and "Off" positions for every 2 deg interval of the machine cycle. There are 8 monitors for external control circuits and Londex relays are supplied as standard equipment. Other types of relays can be fitted to meet special requirements. Further monitoring circuits can be added to a control unit at any time. The input voltage may be either 230V or 110V. Separate tappings are provided on the transformer, which gives an output of 70V, A.C., to a germanium-type rectifier giving an output of 50V, D.C., for the sequencing switch and control system. The input voltage is connected directly to the external circuits and the indicator lamps.

External circuits are connected to the control system by special Niphon plugs and sockets. These are of the offset type to ensure correct phasing if the external circuits are fed from another source of supply. An extra plug and socket is provided, connected to a dummy selector socket, to enable an external interlocking switch to be connected in series with the "Off" position at, say, the T.D.C. of a press crank. This makes it possible to control an operation after the sequencing switch has come to rest. Typical use is to control transfer mechanism or a conveyor running from the press.

A dust proof casing 12 in × 12 in × 6 in deep houses the sequence switch, Figs. 3 and 4. The only moving part of

the switch is the contact arm, which is mounted on a $\frac{3}{4}$ in diameter shaft running in sealed ball bearings. Three brushes are carried on the rotating arm; two make contact with the segments of the 9 in diameter stationary commutator and one with the annular central contact. The control panel, Figs. 2 and 7, is also enclosed in a dustproof casing, the overall dimensions of which are 34 in $\times$ 18 in $\times$ $7\frac{1}{4}$ in deep. This has a hinged access door furnished with a lock.

The other model, Type 72/6, is of the same general design but is arranged with only 72 sockets on the panel and a 72-segment commutator in the sequence switch to give "On" and "Off" positions at 5 deg intervals of the machine cycle. It is normally equipped for the control of six external circuits.

In general, the equipment is simple in design, construction, and operation, and is readily understood by electricians and machine and tool maintenance personnel. The principle of the commutator and brush is well proved and in the case of the sequence switch the speed of operation is low as compared to that of electric motors or generators. Maintenance is claimed to be negligible. In a test run of one million operations only one replacement was necessitated; a visual indicator lamp.

Operating advantages

An actual example of the Masterotor equipment in use may be cited from the press shop at Briggs Motor Bodies Ltd., of Dagenham.* There it is used to control a double-acting press at the start of a press line producing car door panels. The press is fitted with an automatic blank loader at the front and an automatic unloader at the back, with a turn-over device delivering the inverted panel to a conveyor to the next press in the line. Built into the press tools are pneumatically operated locating gauges, lifters, ejectors, and also spray lubrication.

The blank is fed in automatically against the locating gauges and is sprayed with lubricant. Gauges are withdrawn during the down stroke. On the upstroke, the formed panel is automatically lifted and ejected from the die and gripped by the jaw of the unloader. As the panel is being drawn out the locating gauges are lifted to position ready to receive the

*See "Mechanization of Body Production," *Automobile Engineer*, Feb. 1957.

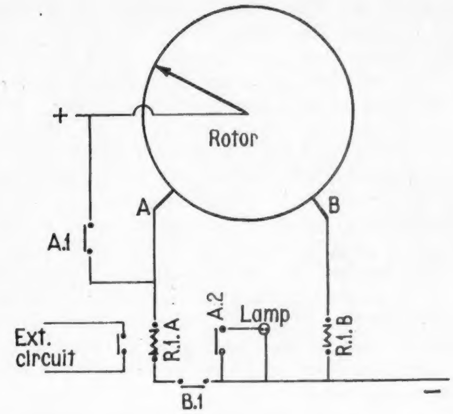
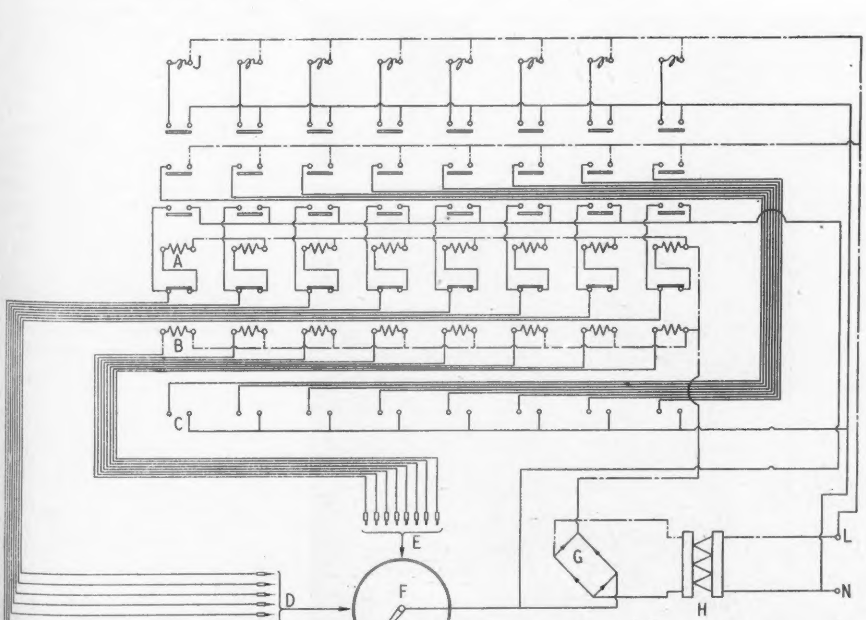


Fig. 5. Circuit diagram of monitor. One monitor is provided for each control circuit

incoming blank. All these automatic operations are initiated by solenoid-operated valves controlled by the Masterotor equipment.

The major advantage of this method of control was the elimination of the "dwell" experienced with conventional cam-operated switch control. This enabled every operation to be non-overlapping. The operation of various functions in "cascade" sequence is of value as it facilitates the location of any mechanical failure in the various items of handling equipment. The more precise accuracy of setting made possible by the Masterotor selector panel allowed the blank to be fed so closely behind the ejected panel that it became possible to run the press continuously, instead of intermittently, with a resulting increase in the rate of output. Since the press clutch remains in constant engagement, wear on the clutch, and the necessary maintenance, are substantially reduced.

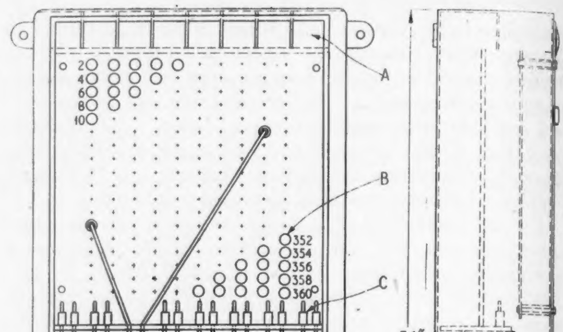
It is, of course, not possible to design sets of tools for a range of differing components so that the automatic operations will occur at exactly the same points in the machine cycle. This arises from the varied depth of draw and similar



A selector relays, 3 N.O. contacts; B selector relays, 1 N.C. contact; C external circuit sockets; D "On" leads, red; E "Off" leads, black; F rotor; G rectifier 50V, D.C.; H transformer 110V-70V; J indicator lamps; L line; N neutral

Fig. 6. Circuit diagram for eight-circuit Masterotor control

considerations and thus, when changing tools, it becomes necessary to re-set the timing of the handling equipment. With the conventional cam-operated switches this was a lengthy, time-wasting procedure and usually necessitated several trial runs before a setting could be proved. With the Masterotor control an initial setting can be effected in a matter of minutes and any adjustments found necessary or desirable can be made immediately in precise increments of 2 deg of the cycle. In the case of repeat settings a trial set-up can be made in a matter of seconds either by means of a setting record card, as mentioned earlier, or by a punched card for the particular job. Such a punched card is suspended in front of the socket panel and the plugs are inserted through the holes in the card to give an accurate setting. Individual idiosyncrasies due to tool setting or to press conditions can be taken care of by the setter, by advancing or retarding the plugs on the panel.



A *visual sequence indicators;* **B** *insulated sockets;* **C** *"On" and "Off" selector plugs;* **D** *selector relays, enclosed by Perspex panel;* **E** *cable counterweights;* **F** *D-P external circuit sockets*

Fig. 7. Standard eight-circuit panel for press controller

Special control units

Standard units are designed for attachment to existing presses or tools, as shown in Fig. 8. The sequence switch is attached at some suitable position to receive a drive either by shaft or chain, and the control panel is mounted either on the press or a nearby support where it is conveniently located for setting operations. In addition to the standard units, however, specially designed or adapted units are produced to be built into either standard presses and machines or in special-purpose machinery.

An example of such a modified unit is shown in Fig. 9. This is a special control unit standardized for new Clearing presses. The press makers have provided suitable space in the press frame and a vertical shaft to drive the contact arm of the sequence switch which is mounted immediately above the control panel. The combined panel and switch unit is arranged as a "slide-in" control unit, interchangeable for top-drive, under-cranked, and two-speed clutch presses. When two-speed clutches are employed, a fast approach speed and retracting speed is used in conjunction with a slower crankshaft speed during the actual drawing of the work. The changeover points, from fast to slow speed on the down stroke and from slow to fast speed on the upstroke, are dependent upon the depth of draw. The Masterotor control provides an easy and convenient method of selecting or adjusting the position in the press cycle of these two points.

Inching

A development of considerable interest in connection with the control of press clutches by the Masterotor system is a positive inching device. The commonly used type of "Inch" button is, in fact, a continuous-run button which

Fig. 8. Masterotor unit mounted on Clearing press and controlling, with other functions, the operation of the automatic take-off device

maintains engagement of the clutch as long as it remains depressed. Both inadvertent operation and sticking of the button have been the cause of serious injury to toolsetters and damage to tools.

With the system evolved in conjunction with the Mastero-tor, Fig. 11, depression of the inching button will result in the press crankshaft moving through only 8 degrees of a revolution. The circuit is then cut, and the inching button must be released and re-operated to initiate a further movement of 8 degrees. Small movements shorter than 8 degrees can, of course, be obtained by releasing the button in the usual manner. Since the inching system is controlled by the angular movement of the crankshaft it is not influenced by factors of time or speed. The complete system is self-checking, and the failure of any component item will result in non-operation.

TOOL PROTECTION

The most common source of trouble experienced in press operation is failure to eject the pressing. It may remain in the die or stick to the top tool with the result that on the next stroke there are two thicknesses of metal in the dies and the tools are damaged or smashed. The cost of tool repairs or replacement and the losses occasioned by down time are likely to be severe. Even where presses are hand loaded such accidents are liable to occur.

Mechanization of loading and unloading will not eliminate the hazard and an operator will be required to watch the functioning of the press and equipment and, particularly, to stop the press in the event of a mechanical failure.

Ejection check

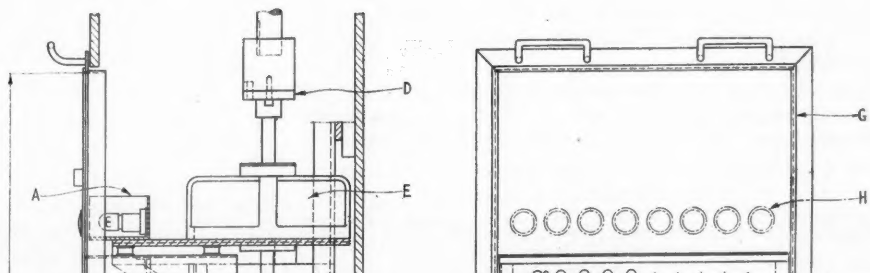
To stop a press automatically in the event of non-ejection of a pressing, and thus to make it possible for one operator to overlook the functioning of three or four presses, the Company has developed a standardized photo-electric unit that can be fitted to any press. This will check the ejection of the work and automatically disengage the clutch in the event of malfunction, thereby safeguarding the tools.

A feature of the electrical control circuit is that the light beam must be broken and remade before the press will re-stroke. This is of importance where relatively long components are being produced, which could be partially ejected, to an extent sufficient to break the beam, but still leave part between the tools. It is important that the beam is projected as closely to the tools as is practical to enable the "clear" signal to be given as early as possible in the press cycle. The device can be used with presses of the friction clutch or key clutch types.

Quick stopping of A.C. motors

On machine tools or special-purpose machines that are not equipped with a clutch mechanism but are directly driven by an electric motor, the need arises for means to effect a quick stopping of the machine in any emergency that may threaten danger to either operator, or tools, or both. Where A.C. driving motors are used and the machine in operation has no substantial store of kinetic energy, as in a heavy flywheel or other rotating mass, a quick stop can be made by means of the I.G.E. D.C.-injection braking unit.

This standardized unit can be wired into "Direct-on" or "Star-delta" starters and is controlled by either a remote switch or a photo-electric device. On receipt of a signal from the switch or from the safety device, the main A.C. current is immediately cut off from the motor. D.C. current is injected into the field winding of the motor, which is thereby converted to act as a powerful electro-magnetic



A visual indicator assembly; B hinged Perspex safety panel; C Perspex panel enclosing relays; D switch drive coupling; E switch assembly; F cable counter-weights; G hinged and locked door; H visual indicator covers; J insulated sockets; K "On" and "Off" selector plugs; L interlock circuit plug; M selector relays, 1 N.C. contact 50V, D.C.; N slector relays, 3 N.O. contacts, 50V, D.C.; P to rear sockets; Q inter-lock circuit socket; R D-P external circuit sockets

Fig. 9. Complete eight-circuit control unit, as built in new Clearing presses

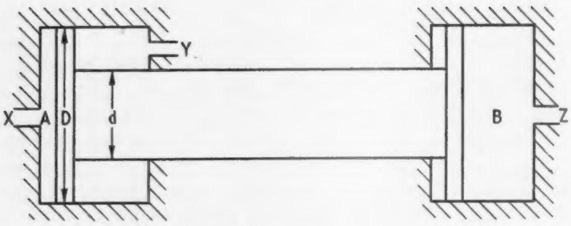


Fig. 10. Spool of special pilot-operated air valve for safety control. In any circumstance it will fail to the exhaust position

brake. As soon as the motor comes to rest the D.C. current is automatically switched off. Both the field current and the timing arrangement are readily adjustable to meet specific conditions.

Safety control of pilot-operated air valves

The wide use of solenoid-operated air valves in automatic actuation and control systems, and particularly press clutch control, has led to the demand for a valve that will, under all conditions, fail to safety and thus preclude repeat strokes. To meet this requirement, the Company developed a monitoring system which, employed in conjunction with a valve specially produced for the Company by Baldwin Instruments Ltd., prevents the possibility of failure to danger and uncovenanted repeat strokes liable to involve accident to operators or tooling.

The first requisite was a pilot-operated valve that always had forces to move the spool to the exhaust position that were of greater magnitude than the forces available to move it to the pressure position. Furthermore, all reliance on springs was to be eliminated. This was achieved by applying constant air pressure on the return side, in conjunction with a duplex pilot system. The valve spool is shown diagrammatically in Fig. 10.

At each end there is a solenoid-operated pilot system, designated respectively X and Z. These are electrically connected in parallel but X system admits pressure air to cylinder A when energized, while Z system admits pressure air to cylinder B when de-energized. At Y is a port continuously open to air line pressure which exerts a constant force on the rear face of the piston in cylinder A. On energizing X and Z solenoids, air is exhausted from cylinder B and pressure is applied to cylinder A, causing the spool to move over to the operating position and to pass mains air. On de-energizing the solenoids, cylinder A is opened to exhaust and pressure air is admitted to cylinder B which, in conjunction with the constant air pressure at Y, forces the spool to the exhaust position, shutting off mains air and exhausting the system.

To examine the main features of this arrangement; if the area of each piston head is designated D, the cross-sectional area of the spool d, and the line pressure available as P, then the effective force moving the main spool to the pressure position is PD—(PD—Pd) since there is a constant force at line pressure on (D—d) at the rear of piston A. Areas D and d are so arranged that D—d = 2d, which means that at all times there is twice the force available to return the spool to the exhaust position than to the operating position, that is:

$$\text{Operating force} = PD-(PD-Pd) = Pd$$

$$\text{Return force} = P(D-d) = 2Pd$$

It is obvious, therefore, that at any particular time, if the condition of friction or striction on the main spool is such that force Pd can move into the pressure position, force 2Pd will return it to the exhaust position. Normally force PD on piston B is available to assist this return, so that under normal conditions there is available a force PD+2Pd to return the spool.

It will be evident that the valve will continue to function quite safely even when any of the following faults may develop:—

1. The operating pilot system sticks open and fails to exhaust, maintaining constant pressure on A. The balance of forces are then $PD < PD+2Pd$, therefore the spool will return.

2. The return pilot system fails and no pressure is exerted on B when the solenoids are de-energized. The spool is returned by constant pressure 2Pd.

Fig. 11. Circuit for inching control from rotary switch. R1 coil energizes the press clutch

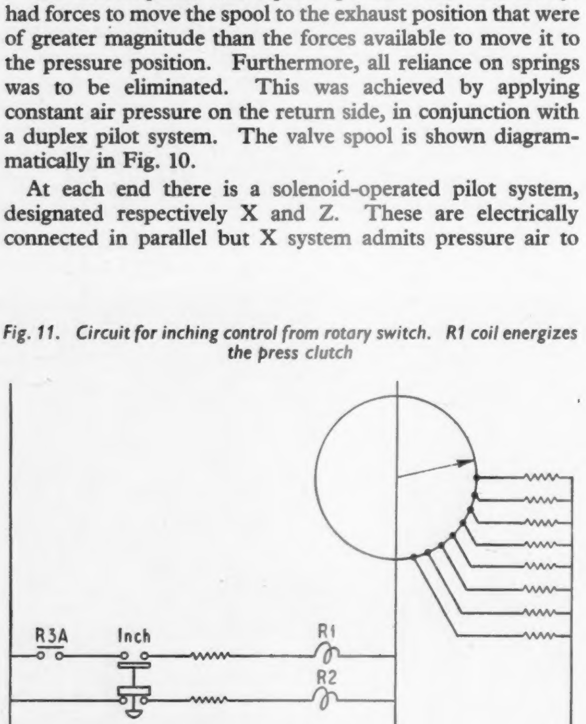
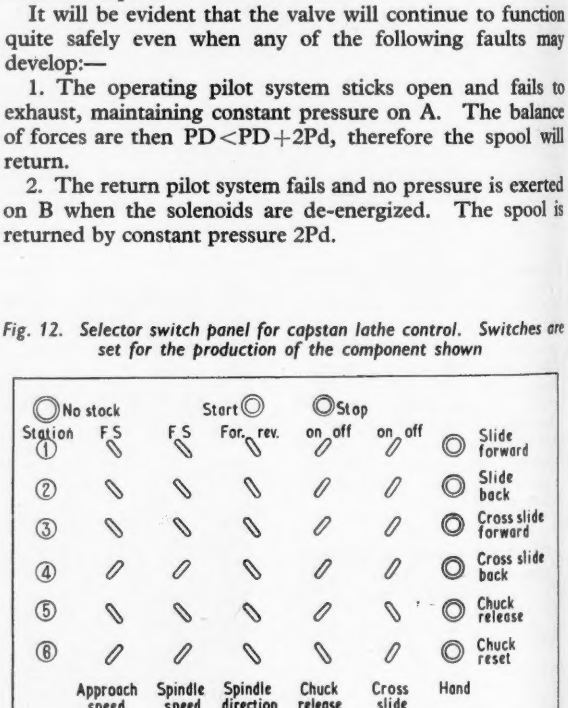


Fig. 12. Selector switch panel for capstan lathe control. Switches are set for the production of the component shown



3. Excessive pressure on spool due to damaged seals or contaminated air supply.

As, however, the valve will continue to function after a fault has developed, unless strict inspection and preventive maintenance is practiced, a combination of faults could eventually cause a failure to danger. The object of the monitoring system is to effect a strict check of each function of the valve during each operation, so that should a fault develop a signal is received and the valve cannot be operated, after attaining the exhaust position, until the fault has been rectified.

The device is, in fact, a negative feed-back system, being completely self-checking so that any electrical failure within its own system will effect valve safety. The monitoring system is housed in an enclosed panel that can be mounted remote from the actual valve; electrical and pneumatic connections are made by plug-in and bulkhead connections. This complete and monitored valve system is the subject of a patent.

OTHER APPLICATIONS

Although the Masterotor system was initially developed for the control of automated tooling on power presses, the principle of the commutator-type sequence switch can be applied to many other machine tools, to special-purpose machinery, or to industrial processes. For electronic systems of control a multi-contact commutator—up to 1,000 segments can be provided—and a binary system of notation are used. The angular positions of shafts or spindles and the linear displacements of tool slides or other moving parts can be accurately translated to electrical impulses without ambiguity.

Capstan lathe control

A typical example is the conversion of a Murad capstan lathe to fully automatic operation. The layout is shown diagrammatically in Fig. 14, and actuation is by pneumatic equipment. Preselection of the various controls is provided by a bank of rotary switches on the panel of a control console. Fig. 12 shows the panel with the switches set to produce the representative component illustrated. The lathe is set in the usual manner with fixed stops. A complete cycle of operations can be followed by reference to the layout, Fig. 14, and the circuit diagram, Fig. 13. In this diagram the Masterotor switch, for the sake of clarity, appears five times although, of course, only a single rotor is used. Where a single selector switch is shown, actually there are six switches; one for each station on the turret. Additionally, each operation can be controlled by push button and each operation is checked by an indicator lamp. These items have been omitted from the diagram to avoid overcrowding.

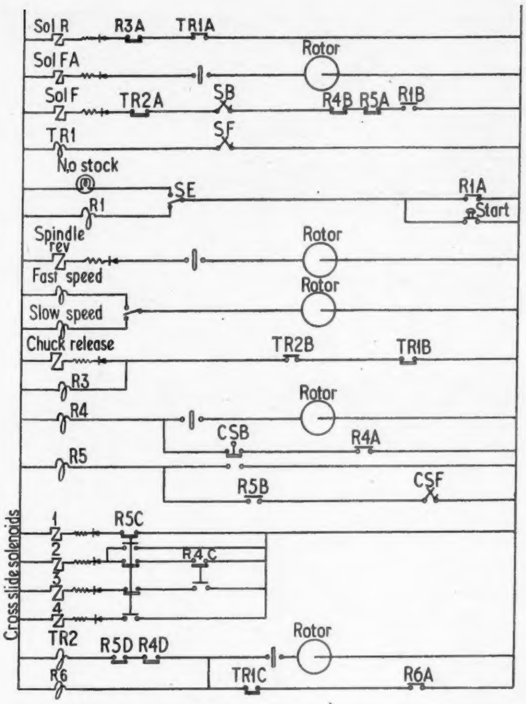


Fig. 13. Circuit diagram for capstan lathe automatic control. Manual control by push buttons, also provided, is omitted from the diagram

Cycle of operations

When the turret slide is fully back the switch SB is made. On pressing the "Start" button, relay R1 is made and is held on by contact R1A. Contact R1B closes, energizing the solenoid valve, and the turret slide is then moved forward. The speed of approach, spindle speed, and direction of spindle rotation have previously been preselected for each station on the control panel. On attaining the fully forward position switch SF is made, energizing relay TR1. This is a time-delay relay ensuring a "dwell" against the fixed stops. Contact TR1A closes to energize the solenoid valve and return the turret slide. The turret is indexed mechanically, simultaneously operating the sequence rotor. On the slide attaining the rear position the switch SB is remade and the turret slide moves forward again on its preselected routine.

This action will be repeated until the station is reached, where the preselected programme demands operation of the cross slide. The switches CSB and CSF are as shown on the circuit diagram when the cross slide is in the central position. Relay R4 is energized during the indexing of the turret and held on by contact R4A. The operation of relay R4 energizes solenoids 1 and 3. As shown on the diagram of the air layout on the cross slide, this causes a backward movement of the slide. At the end of the cross slide traverse,

A solenoid valves (1 and 2 open, centre, 1 and 3 open, back, 2 and 4 open, forward); B spindle reverse; C I.G.E. electromagnetic collet chuck; D cross slide; E restricted exhaust F free exhaust

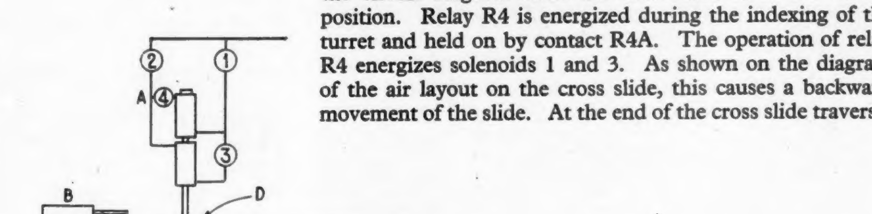
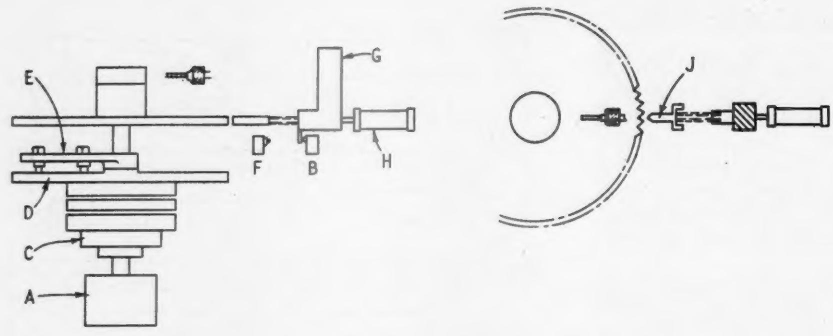


Fig. 14. Diagrammatic layout of capstan lathe for automatic operation



A geared drive; **B** reversing switch **C** electromagnetic clutch and brake unit; **D** commutator; **E** rotor arm **F** switch; **G** drill holder; **H** air cylinder **J** mechanical locator

Fig. 15. Schematic arrangement of angular indexing control

switch CSB is changed over, de-energizing relay R4 and energizing relay R5. This energizes solenoids 2 and 4 on the valve layout, causing forward movement of the cross slide. At the limit of the cross slide forward travel, switch CSF is broken, energizing R5 and leaving solenoids 1 and 2 energized, which return the cross slide to its central position.

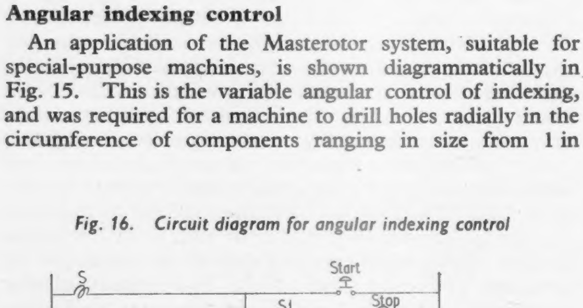
Should only one of these cross slide operations be required the limit switches are adjusted accordingly and the slide will make a small movement in the direction not required before carrying on to the next operation. If chuck operation is also required at the same index station as the cross slide, the operation follows as soon as relays R5 and R4 are de-energized. The closing of relay TR2 causes the chuck to open after the turret with its located stop has moved forward. When the turret has reached its forward position, switch SF is broken and relay TR1 is energized, closing the chuck and de-energizing relay TR2. Contact TR1A is made, reversing the valve, and the slide recommences full cycle.

The delay on TR1 gives a "dwell" to ensure that the slide remains against the positive stops and also for the correct positioning of the stock bar by the locating stop on the turret. Should chuck operation be pre-selected on a different station to cross-slide operation, relays R5 and R4 being de-energized, relay TR2 will be energized immediately. When the stock is exhausted, switch SE is operated, breaking relay R1 and putting on a signal lamp.

Angular indexing control

An application of the Masterotor system, suitable for special-purpose machines, is shown diagrammatically in Fig. 15. This is the variable angular control of indexing, and was required for a machine to drill holes radially in the circumference of components ranging in size from 1 in diameter to $2\frac{1}{2}$ in diameter. The varied diameters necessitated different angular pitchings of the holes. Conventional mechanical indexing devices were rejected as the volume of production was insufficient to warrant a special machine or a special equipment for each specific size of component. By the use of the Masterotor sequence switch it is possible to select the required angular indexing in a few seconds at each change of run.

Fig. 16. Circuit diagram for angular indexing control



Operational sequence

On pressing the "Start" button, relay S is energized and held on by contacts S1. Contact S2 (see circuit diagram Fig. 16), energizes the valve solenoids and the drills are moved inwardly to engage mechanical location and then, with further movement, to drill the component. When the drills leave the rear position switches B are reversed, relay R3 is de-energized and contact R3A makes. The bias on the thyratron grid being negative, relay R4 is not energized. Relay R2 is energized and contacts R2B close. When the drills reach their forward limit, switches F are made, energizing relays R1 and TR1, the latter having time delay for "dwell."

Contact R1A then opens, de-energizing the air valve and the drills retract, relay R1 being maintained by contacts R1B. After the drills have been withdrawn from the component, contact TR1A closes and energizes the clutch. The withdrawal of drills is checked and slowed down pneumatically between switches F breaking and switches B changing over in order to give time for indexing.

The worktable rotates until the next selected contact on the sequence rotor is engaged, when the grid potential is made positive and relay R4 is energized. Contacts R4A and R4B break, the clutch is disengaged, and relays R2 and R1 are de-energized. Switches B are then changed over by the drills attaining the rear position and contacts R3B closed to recommence the cycle. Contacts R3A open to reset the valve circuit.

Rubber in Heavy Engineering

A BOOKLET entitled "The Use of Rubber in Heavy Engineering," by S. W. Marsh, M.I.Loco.E., has been published by the Andre Rubber Co. Ltd., of the Kingston Bypass, Surbiton, Surrey. The contents were first presented as a paper at the Conference on Rubber in Engineering, organized by the Natural Rubber Development Board, on the 27th September, 1956. This booklet contains 54 pages and is well illustrated. It is divided into four sections, which deal with ships and jetties; general engineering, including buildings; couplings; and railway engineering. Sections dealing with the mounting of heavy machines and with railway carriage suspension systems are likely to be of interest to automobile engineers.

FUELS AND LUBRICANTS

Part I: Some Considerations Affecting Their Selection and Use for Road Transport

C. G. Tresidder*

Motor spirit, or petrol, consists principally of hydrocarbons that distil within the temperature range 90 deg F to 425 deg F. It is known that if a fuel is to satisfy operating requirements, certain basic properties must be controlled during its manufacture. In the case of motor spirit, these properties in order of importance are: volatility, octane number, gum content, and contamination by corrosive and other impurities. However, production to satisfy the requirements of commonly established specification limits is not enough to meet the demand for fuels that are competitive in respect of performance. To provide for maximum output, high quality products have some or all of the features discussed below.

Volatility control is important, and it is effected by means of a programme of seasonal refinery blending and the use of additives to balance the need to promote easy engine starting from cold, minimum use of rich mixture during the engine warming-up period, good acceleration, freedom from ice formation inside carburettors, Fig. 1, and the absence of difficulties with regard to hot starting and due to excessive vapour formation in unusually hot weather.

Combustion control is effected by the preparation and selection of high octane value blending-stocks to meet multi-cylinder engine requirements for knock-free operation under road load conditions. In addition, it is desirable to incorporate various ingredients to afford a good measure of protection against pre-ignition, due to glowing deposits in combustion chambers, and spark plug misfiring caused by current leakage across deposits on the centre electrode insulation.

To obtain stability and control of impurities, careful attention must be paid, during refinery operations, to the processing of stocks having good chemical stability. Then, minor quantities of suitable ingredients can be added to prevent gum accumulation in storage, attack on sensitive parts and excessive deposit formation in engines, Fig. 2. Strict distribution routines can also reduce the risk of substandard or contaminated products reaching users' tanks.

Of the many factors that affect the correct burning of a fuel-air mixture in a cylinder, resistance to detonation, commonly referred to as *knock* or *pinking*, is undoubtedly the most significant property of a motor spirit. It is desirable that, from the point of controlled ignition, the compressed fuel-air mixture should burn evenly across the combustion chamber, so that under the pressure of the expanding gases, the piston is forced down smoothly on its working stroke. If the charge burns unevenly, owing to the occurrence of ignition at one or more points ahead of the advancing flame front, then the power output is limited by knock which, if severe, can be serious in other respects. In the laboratory and on the road, the resistance of a fuel to knock is measured in terms of octane number, determined by using blends of known reference fuels. The method of test is important. One is the Research Method of operating the single-cylinder laboratory engine, to simulate low road-speed conditions, Fig. 3. Another is the Motor Method, which is more severe and was commonly used before the war. However, this method is of less significance today, mainly because of

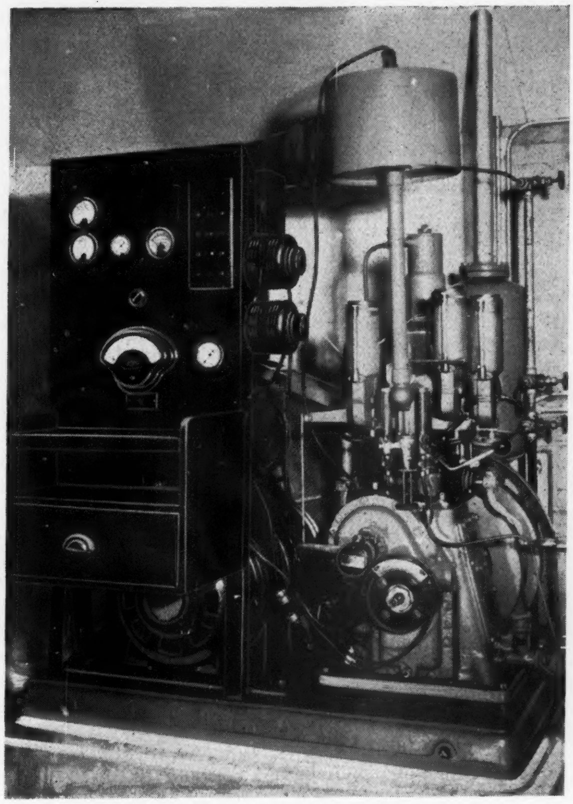
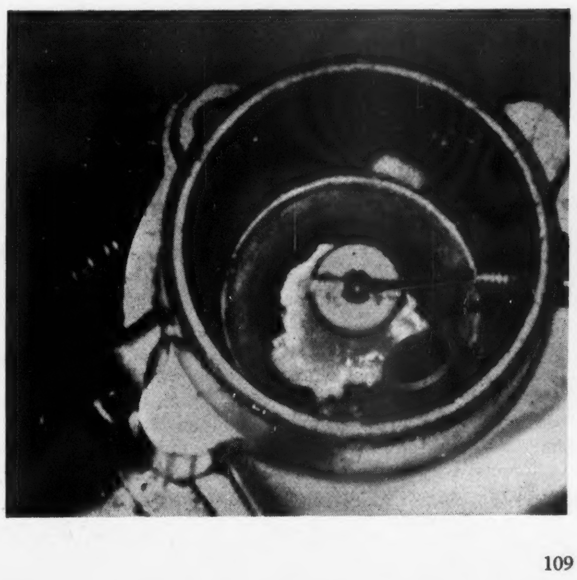


Fig. 3. CFR test-engine for determining fuel octane numbers

Fig. 1. Illustration showing ice formation inside a carburettor



**Chief Automotive Engineer, Mobil Oil Co. Ltd. This article is prepared from a paper presented in London, on the 17th January, 1957, at the National Meeting of the Institute of Road Transport Engineers (Incorporated).*

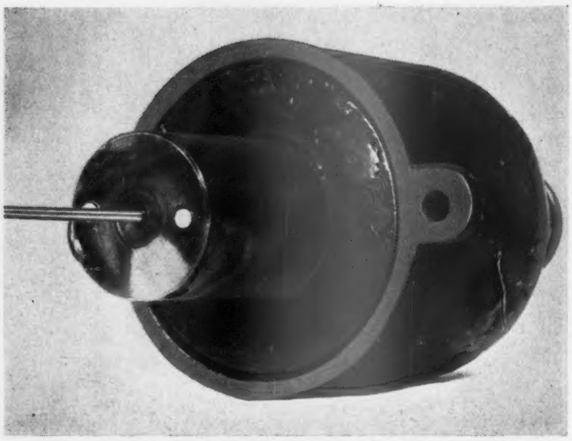


Fig. 2. Example of excessive deposits on carburettor components

marked changes in modern fuel and engine characteristics.

To assist in economically providing good anti-knock performance on the road, a complex lead compound is added in controlled quantities to all motor spirit in Britain. In the past, this additive was associated with reduced exhaust valve life, particularly in some engines of early design, and where inadequate valve metals or inferior repair or maintenance practices were used. Design, installation and maintenance aids to provide for acceptable exhaust valve life with leaded fuels are now more widely understood.

Operators' problems

Spark ignition engines may knock for any one of a number of reasons or for a combination of reasons, such as too high a compression ratio for the fuel used, incorrectly adjusted ignition timing, over-heating of the engine owing to undesirably weak fuel-air mixture, excessive build-up of combustion chamber deposits, and the manner in which the vehicle is used and maintained.

With a mixed fleet consisting of light trucks, utility vans and passenger cars, the problem of motor spirit quality selection is often complicated because vehicle requirements vary over a very wide range, as illustrated in Fig. 4. Where it is desirable, from the point of view of economy, to use only one quality, and this is lower than the top requirement, attention and adjustments become necessary to obtain the best overall result. Fleet experience soon shows which vehicles are more sensitive to fuel quality than others, and usually only a minor percentage is really troublesome.

Some of the principal considerations applying to the correction of critical vehicles are as follows. Each degree of basic spark-timing advance usually increases the anti-knock requirements of an engine by approximately one octane number, Fig. 5. In some instances, ignition distributors leave the factory with production tolerances that give, as between different units, variations in timing equivalent to four octane numbers in engine anti-knock demand. Incorrect control mechanisms and basic settings can alter anti-knock requirements by as much as fifteen octane numbers. In addition, the spark timing for individual cylinders may vary because of worn cam lobes or a distorted distributor shaft.

Surface-ignition can be caused by local hot spots; this, of course, may lead to failure of components, Fig. 6, and requires correction. Modern engines can also be affected badly by combustion chamber deposits that glow readily and cause pre-ignition, Fig. 7. Some modern fuels and motor oils have properties that assist in the control of this problem. Tests for surface-ignition can be made accurately with instruments such as the Cathode Ray Engine Analyser, Fig. 8, but fleet operators can usually carry out check tests quickly by driving the vehicles hard to reach maximum engine temperature and then observing whether the engine continues to operate with the ignition switched off and the throttle in the wide-open position. An unusually high charge temperature, due to the induction hot-spot, valve sticking or restricted cooling, etc., needs to be avoided, as it results in an undesirable increase in engine octane number requirements. In passenger cars and light-duty vehicles engaged on low-speed operations, combustion chamber deposits accumulate relatively quickly and increase the octane number requirement. Employment of an S.A.E. 10W-30 engine oil can reduce this tendency considerably.

Compression ratios vary considerably with different makes, models, and years of vehicle. Engines with high compression ratios, of course, demand higher octane number fuels to provide the improved performance and economy of which they are capable. Fig. 9 represents composite data and indicates that 98 to 100 octane number is required when compression ratios reach 10:1. Excessively lean fuel-air mixtures increase the octane number demand, by promoting over-heating of the engine. Undoubtedly the most important variable within the control of the fuel user is the manner in which vehicles are driven, Fig. 10. Marked improvements in fuel economy are possible with steady driving, as economy runs demonstrate and as every good driver knows.

Derv

The properties of fuels suitable for use in high-speed compression ignition engines depend mainly upon the nature of the crude oils from which they are produced and the refining practices employed. Currently, the products from United Kingdom refineries with which we are principally concerned, distil mostly within the range of 345 deg F to 690 deg F. In Britain, the minor variations in engine design and operating conditions have not so far led to a demand for

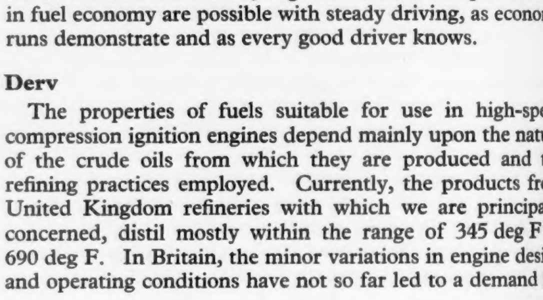


Fig. 4. Diagram showing the effect of variations in octane number requirement for the vehicles of a mixed fleet, each of which has been tuned to obtain the optimum spark timing

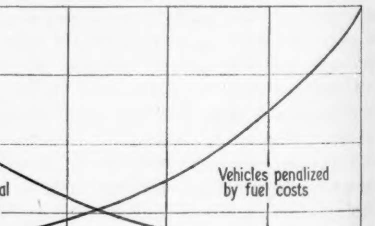


Fig. 5. The effects of ignition timing on individual engine operation and octane number requirement

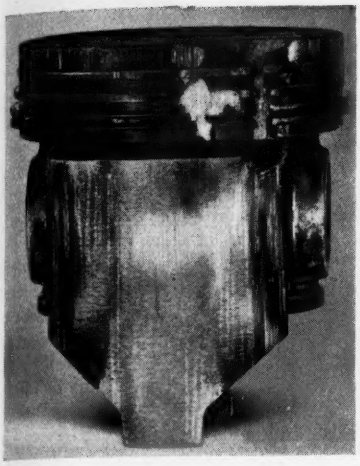
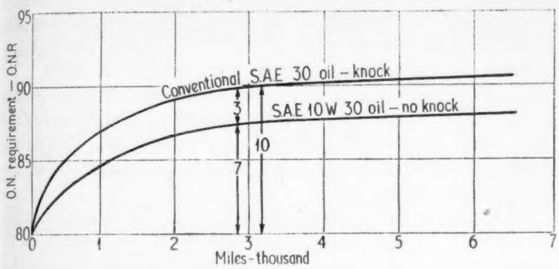


Fig. 6. An example of piston failure due to severe pre-ignition

Fig. 7. Octane number increase due to the build-up of deposits in the engine combustion chambers with two different types of engine lubricating oil



fuels of markedly differing boiling ranges and other properties, as has been the case in the United States of America. It is perhaps of interest to recall that the American Society for Testing Materials have produced two specifications for automotive diesel fuel, one more volatile than the other. In selecting fuels for road diesel engines, operators of large fleets often employ procurement specifications, and it may be helpful to review the various fuel properties with which these specifications could be concerned.

Cetane number is a measure of the ignition quality of a fuel. This factor can influence cold engine starting, light-load operation and the rate of pressure rise in the combustion chamber. It is determined in a laboratory engine by comparing the fuel under test with reference fuels whose ignition qualities are known. One of these reference fuels is cetane, which is rated at 100, and the other is alpha methylnaphthalene, which is given a rating of zero. As in the case of octane number, increases in cetane number over the level required do not improve engine operation or fuel economy. In Britain to-day, the cetane number of the derv available rarely falls below 47, and is often better than 50, which is considered to be adequate.

The distillation or boiling range of fuels is frequently expressed in terms of the temperatures at which 10 per cent, 50 per cent and 90 per cent fuel is recovered, and the final boiling point. The 10 per cent point is an indication of initial volatility which, in certain circumstances, can be related to cold starting and light load operation, while the 90 per cent and final boiling points are a measure of the proportion of heavy fractions present. The extent to which heavy components can be tolerated in diesel fuel depends mainly upon the need to restrain objectionable exhaust smoke, a matter which is to-day regarded as of greater public importance than perhaps it was in the past.

Flash point is not directly related to engine performance, but it is of importance in connection with safety precautions and legal requirements involving fuel storage and handling.

With regard to viscosity, it has been considered advantageous in the past to specify the minimum value, principally because of possible injection pump leakage. However, this does not now appear to be so important, since restriction of the boiling range automatically limits viscosity. The influence of sulphur on engine wear and deposits has attracted considerable attention over the years, and the many papers

Fig. 8. Patented Cathode Ray Analyser for engine investigations

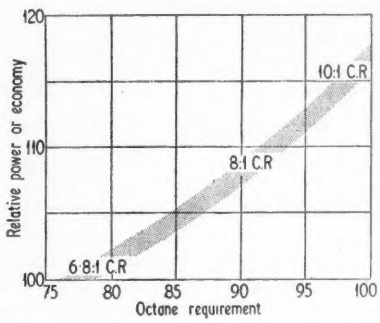


Fig. 9. Gains in power and economy that can be obtained by raising the compression ratio, and the probable requirement with regard to octane number

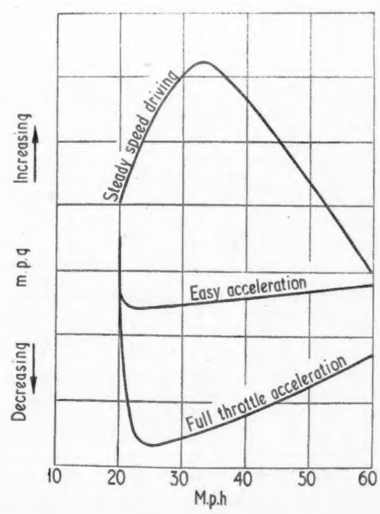


Fig. 10. Illustration showing how fuel consumption varies with different types of vehicle-operation

that have been published point to widely differing effects, which depend mainly on operating conditions and engine design. To provide for maximum availability, B.S.209 limits the total sulphur content to 1·5 per cent but, nevertheless, it is common knowledge that derv in the United Kingdom has a total sulphur content below 1·0 per cent.

Cloud and pour-point determinations indicate respectively the temperatures at which wax formations occur and the fuel solidifies. Both are of importance in connection with the lowest temperatures that fuels can reach before flow difficulties occur. These properties are usually inter-related with cetane number and volatility, and, often, better cold test properties can be obtained only at the expense of increasing the volatility, where this is practical. In Britain, only in severe cold weather do we encounter occasional filter choking due to wax separation. To combat this, the fuel filters should not be allowed to become excessively chilled, particularly if quick engine starting is of paramount importance.

Specific gravity is of some interest, because it can be directly related to the heating value of a fuel. The higher the specific gravity, the more heat units per gallon are available. However, the practical limit to which high specific gravity fuels can be employed depends upon the need for volatility control as a means of assisting in the reduction of objectionable smoke. Also, for fuels with similar boiling ranges, a higher specific gravity usually means a lower cetane number.

Carbon residue is a measure of the carbon depositing tendencies of a diesel fuel when heated in a bulb under prescribed conditions. While it does not directly correlate with the tendency to form deposits in engines, this form of test is considered useful in detecting contamination. Owing to contamination, metallic soaps may also be present, and a test for these ash-forming sources, which contribute to engine wear and deposit formation is also useful. The copper strip corrosion test does not directly correlate with engine corrosion, but it does serve as a means of indicating the possibility of difficulties arising with copper and brass or bronze parts in fuel systems of compression ignition engines.

Diesel smoke

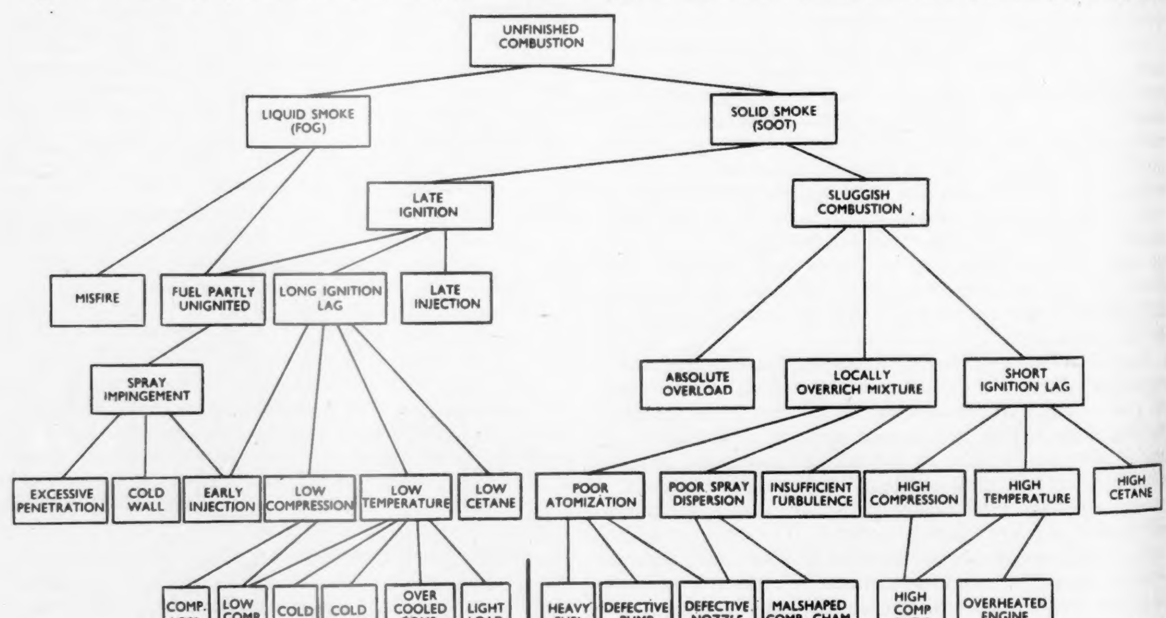
Recently, atmospheric pollution has become a matter of public concern. Legislation has been enacted and further controls are under consideration. With the vast increase in the number of vehicles on British roads, smoke emission from diesel engines has attracted considerable attention. It is of importance, therefore, to consider in some detail road diesel smoke and the steps that can be taken to control what is, at least, a public nuisance.

Diesel smoke can be divided into two classifications: over-load smoke and light-load smoke. Over-load smoke is predominantly black in colour and occurs with any diesel engine if the load is heavy enough, while light-load smoke is predominantly white in colour and occurs mostly during engine starting, idling and acceleration. A summary, by Schweitzer, of many of the factors known to influence diesel smoke is given in Fig. 11. Of fuel properties, volatility is the most significant in its influence on smoke emission. It has been found that smoke can be reduced as end distillation points of fuels are lowered. However, what is of major importance and of particular interest to road diesel engine operators is that engines in good mechanical condition and correctly adjusted will give reasonable freedom from objectionable exhaust smoke on a wide range of fuels. As mechanical condition deteriorates, so engines, when operated on fuels of markedly differing volatility, begin to show differences in smoke emission characteristics. However, these differences tend to even out as mechanical condition deteriorates still further.

All available information seems to be in agreement that the mechanical condition and adjustment of an engine overshadows any other factor or combination of factors relating to exhaust smoke; second in importance comes the adverse influence of low water-jacket temperatures. Finally, it is suggested that operators attracted by a fuel of higher than normal specific gravity should enquire as to its cetane number, volatility range and potential effect with regard to the output of exhaust smoke under their operating conditions.

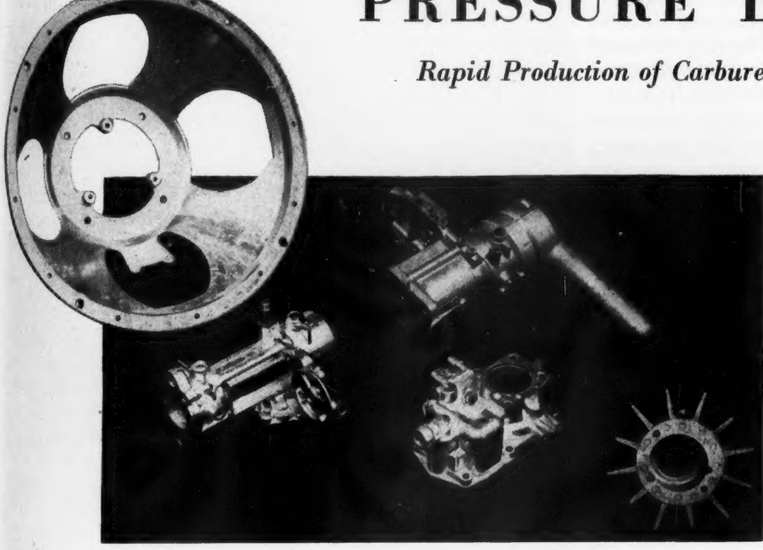
(To be continued)

Fig. 11. A summary, by Schweitzer, of the many factors known to influence the formation of diesel smoke in compression ignition engines



PRESSURE DIE-CASTING

Rapid Production of Carburettor Bodies to Precise Limits



A selection of components produced by pressure die-casting. They are as removed from the machine, except in that the flashes have been trimmed off. A casting runner is still attached to one of them

OF the components produced by the pressure die-casting technique, automobile carburettor bodies are some of the most complex and illustrate well the versatility of the process. The Stanmore Engineering Co. Ltd.,* besides producing the general run of die-castings for other industries, has considerable experience in manufacturing the complex dies required for carburettor components. Moreover, the Company uses these dies in pressure die-casting machines for the quantity production of the final product. All the dies are designed and manufactured on the premises, and trimmed die-castings are the final product. As well as manufacturing the dies, the firm also maintains and modifies where necessary all the casting machines and manufactures any spare parts and replacements required.

Casting has always been a popular method of shaping metal, because material scrap—which can be expensive—is kept to a minimum. This condition is especially true of pressure die-casting. It is estimated that out of every 100 tons of metal received into the raw material stores, 99 tons leave the factory in finished form as castings and only one ton is lost. To achieve this high proportion of material utilization, a comprehensive and efficient method of reclaiming all casting risers and excess material removed in the form of flash is needed.

By the very nature of the process it is necessary that the alloy to be cast should possess a high fluidity at a relatively low melting point, otherwise the life of the die is adversely affected. This requirement necessarily restricts the number of alloys that can be used, and the most popular ones undoubtedly are those with a zinc base. For carburettors a zinc base alloy to B.S.1004A is used. This alloy specification is quoted in detail in the Table; it contains 95·8 per cent zinc, 0·04 per cent magnesium and 4·1 per cent aluminium. Impurities present are iron, lead, cadmium and tin. A similar alloy, identified by the letter B, is also manufactured and has, in addition to the alloying elements already quoted, one per cent copper. This alloy is not, however, generally used for carburettor work.

The features that influence the choice of the type A alloy are that it has a high dimensional stability. This characteristic is maintained even when the part is subjected to heat whilst in service. A feature of the castings produced is their good ductility. The tensile strength of the material exceeds 18 ton/in^2 and it is associated with an elongation of 12 per cent, measured over a 2 in gauge length.

Good pressure die-castings can be produced consistently only if rigorous control is maintained at all stages of the manufacturing process. In this respect pressure die-casting is akin to welding in that corrective or repair action on the final product, as a result of any process discrepancies, is difficult, if not impossible. Care is especially needed in the handling of the molten alloy, as purity and the avoidance of contamination are essential. Zinc originally had a bad reputation as a pressure die-casting material because it was very susceptible to intercrystalline corrosion. With this type of corrosion, the attack occurs around the boundaries of the individual crystals in the metal matrix; a bulky by-product is formed between the grains and causes the part to distort and finally to disintegrate.

After systematic research, it was discovered that this susceptibility to intercrystalline corrosion was only in alloys where the impurities already mentioned, that is iron, lead, cadmium and tin, were present. If these impurities are kept to the extremely low values quoted in the specification, troubles from intercrystalline corrosion are not experienced. Since all possible scrap is reclaimed, it is essential that the recovery system should be absolutely clean in itself so that there is no chance of the deleterious

TABLE—Composition of die-casting alloys to B.S.1004

Material	Content	Alloy A per cent	Alloy B per cent
Aluminium	Minimum Maximum	3·9 4·3	3·9 4·3
Copper	Minimum Maximum	— 0·03	0·75 1·25
Magnesium	Minimum Maximum	0·03 0·06	0·03 0·06
Iron Lead Cadmium Tin	Maximum Maximum Maximum Maximum	0·075 0·003 0·003 0·001	0·075 0·003 0·003 0·001
Zinc	—	Remainder	Remainder

*Stanmore Engineering Co. Ltd., Lowther Road, Stanmore, Middlesex.

impurities being introduced into the remelted stock. For first-degree complexity castings, only virgin metal received from the raw-material supplier is used. For second-degree complexity components the metal used is compounded of 50 per cent reclaimed scrap and 50 per cent virgin alloy.

As has already been mentioned, to obtain consistent dimensional accuracy and good quality castings, strict process control is essential. The four principal measures taken to maintain this control are as follow. A sample of the metal is taken daily for chemical analysis to ensure that the material conforms to specification. The casting temperatures are controlled within narrow limits, and the dies are maintained at their appropriate temperatures. Last, but not least, care is exercised in the design both of the dies and of the provision made in them for cooling. Another factor that has been found to influence the quality of the castings produced is the operator's mode of work.

The temperature cycle is important. Dies are alternately heated by the incoming mass of liquid metal then cooled by water that is constantly circulated through the special channels provided in them.

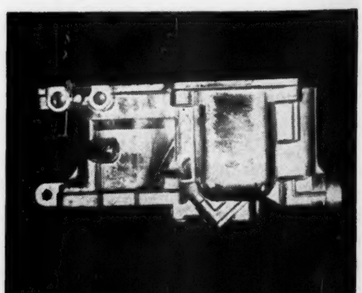
As the casting is chilled and the part is removed, the die-temperature falls. The die is then closed to receive the next molten charge. At the conclusion of the casting cycle the temperature is lowest. This minimum temperature level must not be allowed to become too low otherwise the molten metal will freeze too rapidly on entering the die, and result in imperfect castings. On small runs, the rate at which castings are made is controlled in many cases by the operator. A good operator works rhythmically, at a consistent rate—opening and closing the dies, and initiating the pressure feeding of the metal—and to assist him to maintain this rhythm a large clock face equipped with a second hand has been installed at one end of the shop. The frequency at which the complete production cycle can be operated is high: on small hand-operated machines up to 500 castings per hour can be manufactured, whilst between 100 and 150 larger castings per hour can be produced on the larger-capacity automatic power-operated machines.

A factor that has a great bearing on the maximum rate of production attainable is the design of the dies. The drawing of the required finished part is received from the customers and is first examined to see what the difficulties are in making the part as a pressure die-casting. In fact, there are few geometrical shapes that cannot be made by this method, but of course, many shapes need complex dies. Design changes are suggested to obtain the simplest die that will efficiently produce the required shape. In carburettor work the castings produced are within a tolerance of $\pm 0{\cdot}004$ in and the finish on the casting is so good that surface machining is not necessary.

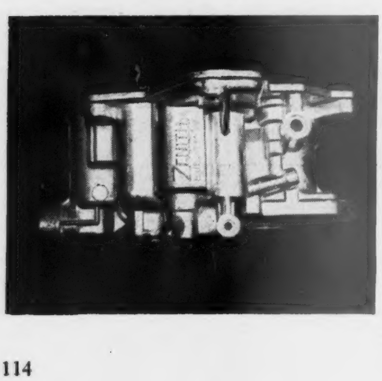
One of the first things to check relating to the final product is that there are no large masses of metal. It is preferable that the casting should consist of a number of thin interconnecting walls of constant thickness and, where large masses have been included in the part designed, it is customary to reduce their volumes by fitting cores. A good carburettor design for pressure die-casting may be considered to be a series of interconnecting membranes or skins that define and enclose the required air passages, petrol chambers and ducts. The external shape is of secondary importance. A disadvantage associated with large masses is that they tend to become porous. In especially bad configurations, there may even be cavities; these cavities form as a result of shrinkage of the fixed volume of fluid metal that enters the die. Since the central mass of the metal remains fluid longest, the cavities tend to form in it. A further advantage derived from fitting cores in the large masses is that the weight of the final component is reduced. Even more important, the volume of material to be cast is reduced and in consequence so also is the cost of production.

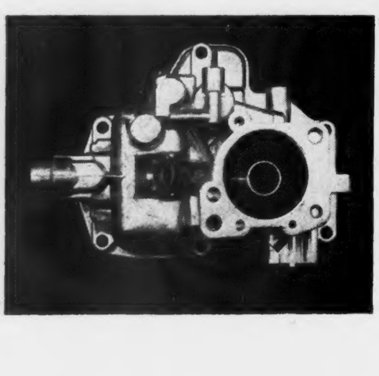
Dies for this type of work consist basically of two major mating sections. The first decision of the die-designer is to select the parting line around the component, that is, the face along which the die halves are to mate. This line does not necessarily have to be in one plane; it is generally defined so that undercuts in either die half are avoided, or at least the number of such undercuts are reduced to a minimum. Forms that are undercut are moulded by supporting the appropriate section of the die on a core-slide. This slide is then moved laterally—at 90 deg to the closing motion of the main halves of the die—and when it is fully closed mates with the remainder of the die. Up to four core-slides can be conveniently fitted around a die but, for extremely complex components, inclined cores operating within the main body of the die may be included.

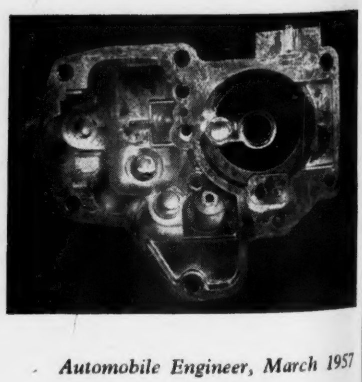
It is also advantageous when designing the die to consider on which machine it is to be used, since the volume of metal that can be pressure-fed at one operation differs according to the capacity of the machine. For very small items it is customary to cut multiple impressions in one die so that several similar components are produced for each

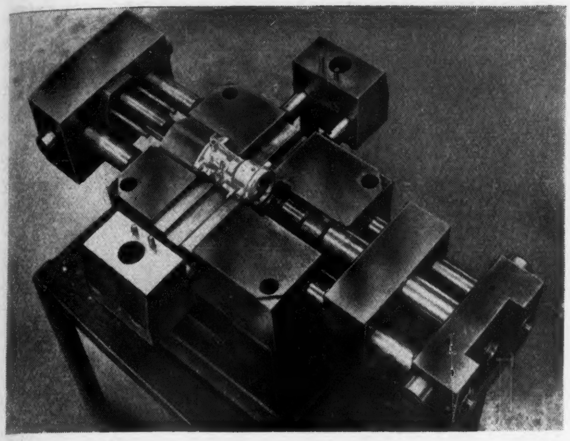


Carburettor body shells are among the more complex components produced by die-casting. Four views of a body shell are shown in these illustrations. The casting has only had the flashes and the runner removed; in other respects it is as ejected from the die-casting machine

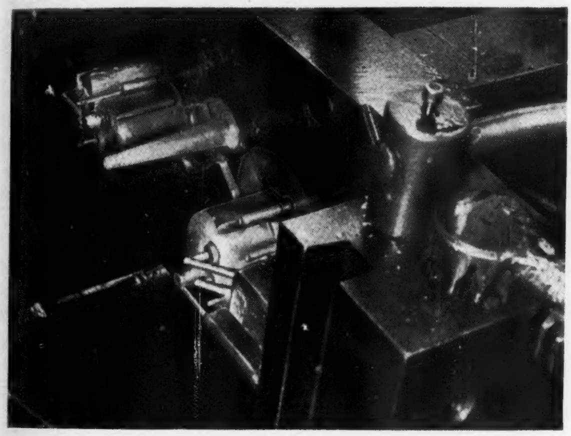








A cast component in a die fitted with two of the four core-slides employed. The rollers on the core slides engage with cam tracks on the machine to move the cores during the opening and closing of the halves of the die



A double-impression die for the casting of carburettor float chambers. The particular application shown in this illustration is a die fitted to one of the larger hand-operated casting machines

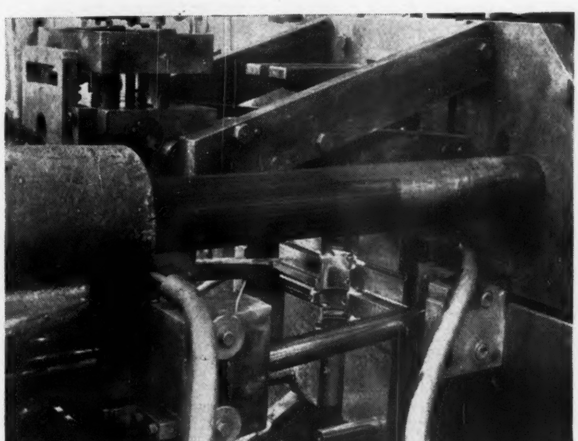
casting operation, or *shot*. It has been found desirable to confine each die to one particular component design because, although there is no mechanical reason against having several different small impressions in the one die, frequently one of the parts has to be modified or it may become obsolete with the consequence that the die has to be withdrawn from production and the balance of production of all the parts concerned is upset. It is also necessary to know on which types of machine the die is to be used; this is because, if any core-slides are required to be moved in the lateral plane, the length of travel has to be determined. Frequently the final die size is determined by the movements required by these core-slides rather than by the size of the product being cast. A rough layout is initially drawn, in which the slide movements are specified and also the position at which the metal is to be fed into the cavity.

Metal is fed through a tapered hole called the sprue. The point at which the metal actually enters the cavity must be carefully chosen, since it will have a great bearing on the quality of the casting produced and on the life of the die. Experience has shown that it is best for the metal to enter the cavity through a wide, shallow lip. The depth of this lip varies between 0·012 in and 0·024 in. The width is not critical, so this lip usually extends around the edge of the component, within the limits to which it can conveniently be incorporated in the design of the die. Liquid metal entering the die is injected through the sprue into a region of the cavity where the flow of the metal is not obstructed by cores. It is also desirable for the initial flow of molten metal to be directed away from cores, which are generally of delicate construction, to prevent them from being burnt and having their operational life reduced. The entry lip is arranged to project the metal so that it flows naturally and smoothly into the shape, in other words, a sharp change in angle of the flow shortly after entry into the die is avoided. Even in large castings, the metal entry is restricted to one orifice because, if two orifices are used, troubles can be caused by eddies where the two streams of molten metal impinge. A sprue giving a thin, wide feed is used because experience has shown that this leads to less porosity in the casting than when other forms of orifice are used.

Another feature of die design that assists in rapid production on the die-casting machine is a high rate of cooling of the molten metal. This is obtained by boring cooling-holes through the die, to pass as near as possible to the large masses in the part. Water is fed through the cooling channel in parallel, that is, the supply is fed directly through many entries on the die. For rapid operation it is desirable that the temperature of the die be kept just below the freezing-point of the metal, but the temperature must not be allowed to become so low that the metal is solidified before it reaches the extremities of the pattern and its finer filamentary detail.

Also included in the design of the die are ejectors that are used to free the final casting from the pattern. In rapid production, the final casting is removed from the die whilst it is still fairly hot; consequently, it is necessary to space the ejectors carefully so that their action does not cause the part to be distorted. Where long thin cores for producing deep holes are fitted, it may be difficult to extract these cores from the body of the metal. To reduce this difficulty some of the ejectors are of hollow tubular pattern to fit round the cores to enable a truly axial pressure to be maintained round the core. A clearance of 0·003 in is allowed between the ejector and the core. These sleeve ejectors also assist in keeping the bore parallel. Where multiple cores of complex shapes are fitted it is customary for them to protrude through what might be termed a stripper-plate, so that as the cores are retracted they do not tend to draw local sections of the casting with them.

The core-slides of the die can be moved by various means. In this application, the lateral slides are withdrawn by inclined pins, while the vertical slides are actuated by a rack and pinion and the motion is regulated by cam tracks with roller followers



The dies, with all their associated core-slides, are sometimes fairly complex mechanical assemblies. There are several methods of drawing cores in laterally as the die is closed and withdrawing them again as the die is opened. Where the core travel is small, inclined spigots can be fitted to one of the main die-halves to engage with holes in the lateral moving core-slide member. On the larger machines hydraulic core-pullers are fitted. These are hydraulic jacks that are operated at the correct moment of the cycle, by sequence switches triggered by limit switches initiated in their turn by movements of the die-halves. Another method of operating the core-slides automatically and in sequence is to have cam tracks that are fixed to the main framework of the machine, in which rollers attached to the core-slides engage. The core-slides are part of the die-half that is opened. As the die-half is retracted the contour of the cam track causes the roller attached to the core-slide to pull the slide away from the die.

Occasionally, where production quantities are small, or where it is inconvenient or impossible to fit automatically operated mechanisms, a rack-and-pinion gear is used to move the core-slide in or out of position. This rack-and-pinion gear assembly is manually operated by a lever. One advantage of hand-operated cores is that a certain amount of feel is obtained and consequently potential trouble can be detected by the operator when he is retracting the core-slides or feeding them into position. Where the core-slides are operated by inclined spigots it has been found that the maximum limit that can be successfully used in practice is 15 deg. With very complex dies incorporating many core-slides that come in from a variety of directions, interlocks are provided to prevent damage to the dies by incorrect sequencing.

If many cores are fitted in one die there is a strict sequence of operation. When the metal solidifies in the die, especially if the shape is complex, the casting tends to grip the cores. The load that would be required if all the die core surfaces were to be broken at the same time is so large that it is doubtful whether some of the machines would have adequate capacity. Furthermore, the hot casting would be subjected to loads that might distort it. To overcome this difficulty it is the custom to withdraw core-slides in sequence so that at any one time only a small portion of the casting is subjected to the breaking loads. For this reason, where inclined spigots are fitted a clearance of 0·08 in is provided in the holes, so that as the die-halves are pulled apart the cores do not begin to withdraw until this 0·08 in clearance has been taken up.

A Schultz pressure die-casting machine that has been modified and equipped with an electronic control system for automatic operation

The raw material is received in palletized form, the ingots being cast in a shape suitable for handling by a fork-lift truck

In the design of the die, it is necessary to provide draft angles in much the same way as with conventional casting or forging practice, so that the component can be removed from the die. The minimum draft angle specified is 0·005 in/inch. Since the component is cast hot, and contracts as it cools, the pattern also has to be made oversize by 0·006 in/inch. This contraction factor is used in all design calculations, irrespective of the mass or geometry of the part to be produced.

The main bodies of the dies are made from a chromium-vanadium steel, supplied in the form of forged blocks or bar. To maintain a good surface finish, sliding portions of the die that could become scored through use are hardened. Other portions of the die that are not subject to mechanical sliding contact are left unhardened. The die is only hardened after it has been proved in the workshop to produce a casting to the required specification. For final acceptance of the die, a practical test is always carried out on a casting machine.

If the component produced is satisfactory, the die is engraved with the appropriate part numbers and lettering, and the core-slides, where necessary, are hardened. In operation, the dies are lubricated by graphite oil sprayed on by the operator, who normally performs this operation after every 12 shots.

As the die is filled with molten metal the air is expelled. Paradoxically, more trouble is experienced in venting air from simple die-cavities than from complex ones. The reason is that complex dies usually have many parting-faces made up of cores and ejector-sleeves, all of which provide leakage vents for the air. Simple parts where only the two die-halves come together do not have these alternative paths along which air can leak, and sometimes it is necessary to provide additional ones in the die. These vents are made by cutting grooves, 0·005 in to 0·01 in deep, to form gates across the mating faces of the dies. These gates naturally create flashes on the finished component, but they can easily

be broken off when the casting is trimmed. Although the metal is injected into the dies under pressure and these gates are vented to atmosphere there is no danger of the molten metal being sprayed out of the die as the venting path is made sufficiently long so that the thin stream of metal is cooled and solidified before it approaches the outside of the die. Where the volume of air to be released is even greater, a hollow is machined adjacent to the die impression, to which it is connected by a number of shallow gates.

Pressure die-casting machines vary according to the individual design, but the principle of operation is the same. One half of the die is considered to be the fixed portion and the other half is attached to the movable head. When closed, the die-halves are maintained in close contact—against the pressure exerted by the metal—by the closing load imposed by the machine and wedges that are attached to the side of the die and engaged as it is closed. A gooseneck container mates with the sprue-hole of the fixed portion of the die, and it is through this hole that the molten metal is injected into the die. Goosenecks can be fixed or movable according to the particular design of the machine. The gooseneck itself is immersed in the molten metal, which is contained in a small heated pot at the rear of the machine. Pressure is exerted on the metal either by a pump or by compressed air fed into the enclosed volume of the gooseneck to act on the free surface of the metal. The exit from the gooseneck is an inch or so above the general level of the molten metal in the heated pot; at the end of the pressure feeding cycle, the pressure is released and the molten metal falls back from the exit orifice, to leave it clean for the next operation. It is also customary to heat the nozzle of the gooseneck to ensure that no semi-hardened metal remains. The molten metal is fed into the dies at pressures ranging between 450 to 3,500 lb/in².

One of the critical factors over which strict control is maintained is the temperature of the molten alloy in the holding pot of the machine. All the machines are equipped with an automatic means of controlling the casting temperature of the alloy. The holding pot is heated by gas jets, the volume of gas burnt being automatically controlled by a pyrometer that monitors the temperature of the metal in the pot. This temperature is maintained between 410 and 420 deg C.

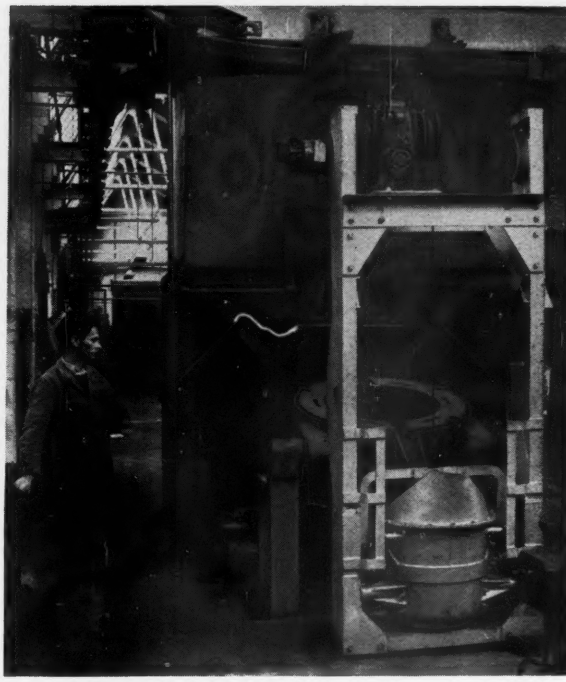
In this particular foundry, six major types of pressure die-casting machine are used. These are the 21 in Schultz hydraulically-operated machines, 1½Z Reid-Prentice hydraulically-operated machines, No. 12 E.M.B. air-operated machines, M.55A manually-operated machines, and two types of Buhler machine, also manually-operated.

The casting capacity of a machine is measured by the maximum weight of metal that can be pressure fed at one time. Small hand-operated M55 machines have a capacity of 6 oz and the larger automatically-operated Schultz units have a capacity of 25 lb. Both weights include the runner to the casting.

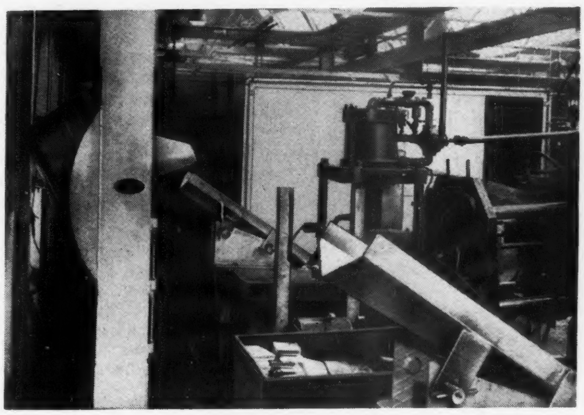
With the larger completely automatic machines, about 1½ days is necessary for the setting up for a production run, but the smaller hand-operated machines can be set up in a matter of hours. For short production runs the versatile Buhler machines are especially useful, because they are hand-operated and consequently can be set up in a short time; they are also of fairly large capacity, being suitable for the production of castings up to 18 lb in weight.

The complete work sequence to produce a die-casting is as follows: virgin metal in palletized form is withdrawn from the raw materials store by fork-lift trucks, and is delivered to 350 kg, oil-fired furnaces. Here the metal is melted down and is subsequently transferred to a travelling ladle. The travelling ladle runs on an overhead track down the rear of the pressure die-casting machines which are arranged in a line along one side of the foundry. This travelling ladle can be elevated by an electric motor and is fitted with cam tracks so that it is at the same time automatically tilted for pouring. The molten metal is received by a chute leading to the heated holding pot of the machine. With certain machines this cannot be conveniently accomplished, so a separate heated bale-out pot is provided at the side of the machine and is filled from the travelling ladle.

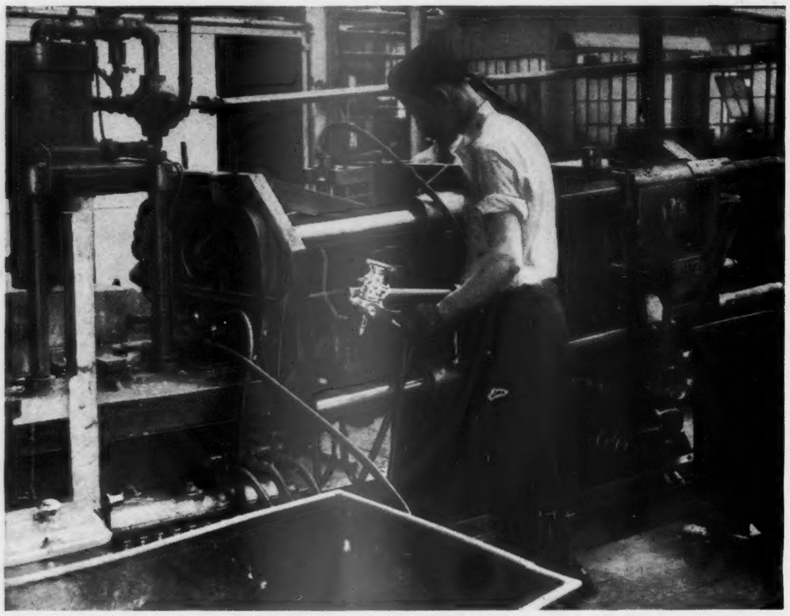
As the machines are operated, the castings are withdrawn from the dies by hand and placed in work-pans. It is a rule with this foundry that the finished casting should be removed by hand, and not just ejected and allowed to fall down a chute; this provides a simple solution to the stacking problem, and avoids damage that otherwise might possibly occur whilst the casting is still hot. The pans are passed across to operators who stand adjacent to a continuous conveyor-belt



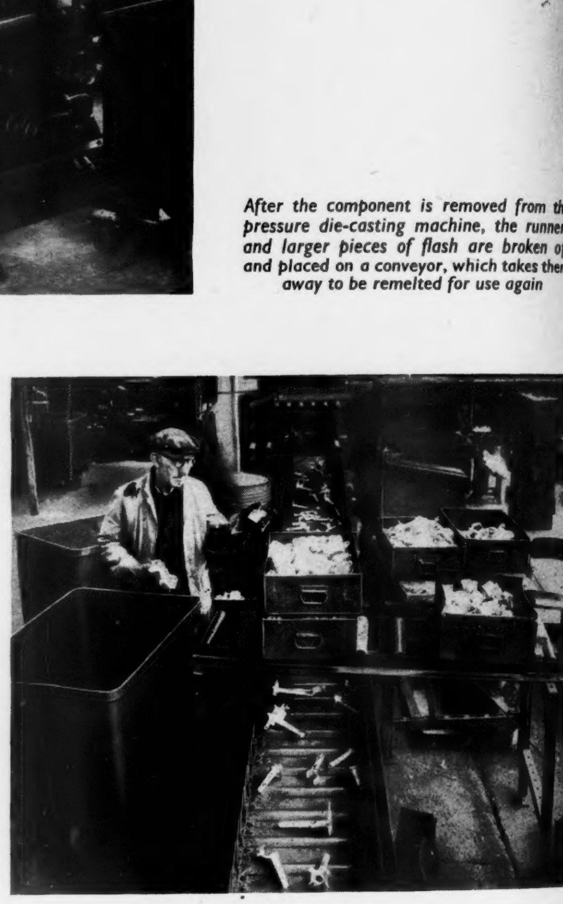
Travelling ladle being filled from the 600 kg melting furnace. In the background is the 20 ton hopper for material to be reclaimed



A travelling ladle moves along the rear of the line of casting machines and discharges into chutes, down which the liquid metal flows into the holding-pot that is incorporated in the machine

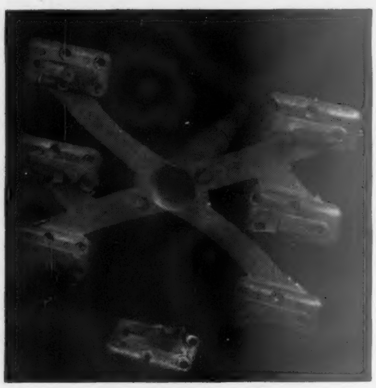


An operator removing by hand a carburettor body that has just been ejected from the die on a Schultz die-casting machine



After the component is removed from the pressure die-casting machine, the runners and larger pieces of flash are broken off and placed on a conveyor, which takes them away to be remelted for use again

that traverses the entire length of the foundry. These operators break off all the excess flash and the runners and drop the castings on to the conveyor. Then the castings are passed by the conveyor to a line of women operators who trim and clean them where appropriate. The flash and runners are conveyed to the end of the shop, dropped on to another conveyor, elevated, and finally loaded into a two-

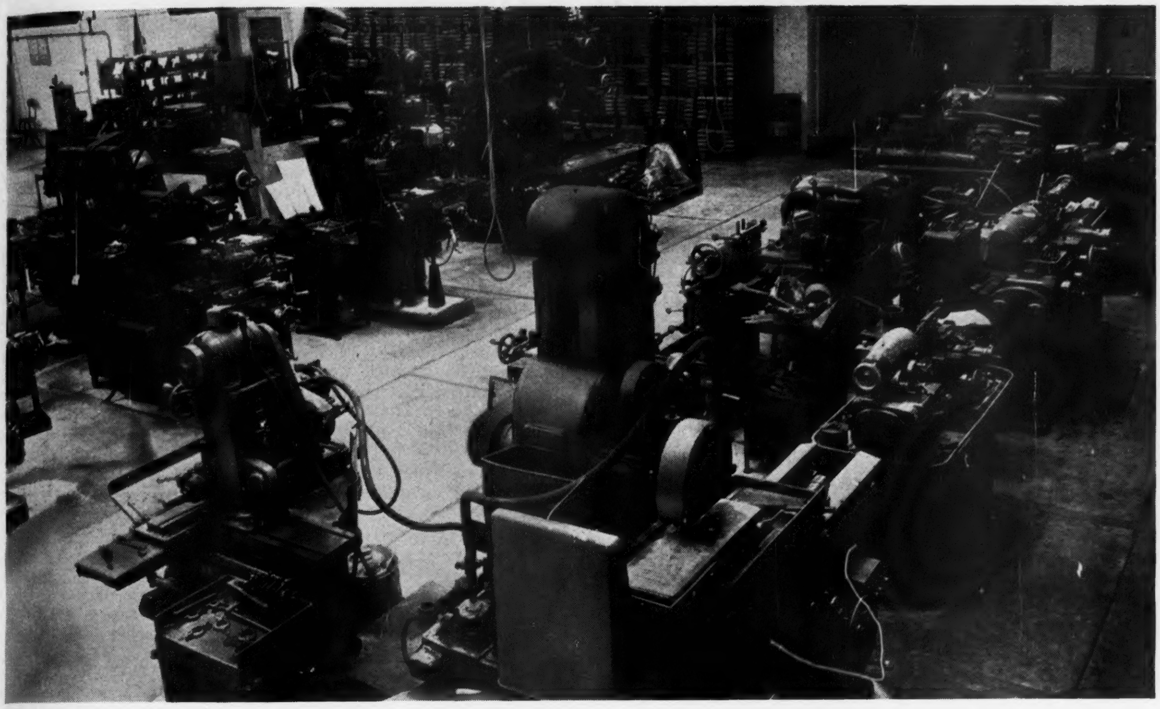


Small components are produced in dies with multiple impressions. These six emulsion-blocks complete with runners are in the condition in which they are removed from the die

compartment 20-ton hopper. From the hopper the segregated scrap can be withdrawn as required and remelted in the 600 kg melting furnace.

Good quality castings are ensured, as has already been mentioned, by strict control of all aspects of the process. For the quantities produced, it is obviously not possible to inspect every individual casting nor is it necessary to do so. But to ensure control of quality, an inspector visits each machine at least once an hour and takes the casting manufactured at the time of his visit. He then returns to the inspection department where all the critical dimensions are checked against a master casting. Complex castings are sectioned to ensure that long small-diameter cored holes are straight and true and are correctly centred. If any error is discovered, such as might be caused by a broken core, then the die-casting machine is withdrawn from service immediately and all castings manufactured in the period between the two inspections are segregated. If the particular component being produced is known to be especially difficult or troublesome, the inspector visits the machine twice an hour during the production run. Where no defects are discovered, the test castings are kept for 24 hours. When the day's production has been packed and dispatched, the test items are then returned on the conveyor to the two-compartment scrap hopper.

A further inspection technique used to provide a rough guide as to whether porosity or voids are present is to weigh the casting. The value of this check varies according to the size of the component: the larger the casting the less sensitive is the check, but for many small components that weigh an ounce or fraction of an ounce the test is reliable and all



The tool room, where all the dies and their components, and the replacements for the casting machines are manufactured

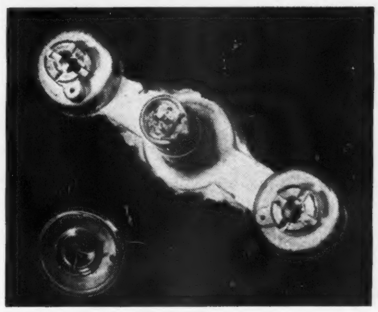
underweight castings are rejected. The minimum acceptable weight for a particular casting is determined from experience and is agreed, between the foundry and the customers, prior to the first production run.

At the beginning of each day, a sample of the molten metal being used is taken at random from one of the storage-pots of a casting machine and is despatched for chemical analysis. The results of this analysis are known within 24 hours and no parts are released from the works until a clean inspection sheet has been obtained. This rule is strictly observed.

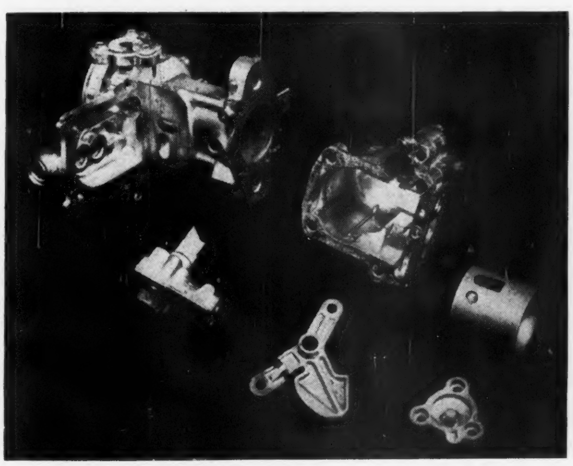
The entire works is laid out for flow production and, although a diversity of patterns may be produced, the nature of the work does not vary greatly and consequently the trimming lines do not have to be constantly modified. At the conclusion of a production run the die is removed from the machine and is sent to a maintenance section, where it is cleaned and oiled and an inspection is carried out to ensure that no parts have been broken or become distorted while it has been in use.

Long thin cores are frequently broken under the harsh conditions to which they are subject, and the servicing section can replace these parts. For this purpose, a stock of over 2,000 standard cores is carried. A long core is defined as one which has a fineness ratio of more than 10:1. Fineness ratios greater than 20:1 are frequently encountered in carburettor work. Cores of this type are subjected to high tensile loads when they are withdrawn from the solidified metal of the casting and, over a period of time, tend to stretch. If, however, the damage is more severe than this, the dies are returned to the toolroom to be repaired. Each die has a history sheet stating all work carried out on it and also giving details of design modifications that have necessitated its being altered. After being oiled and greased the die is then returned to store to await the next production run. The potential operational life of a die may be gauged from the fact that on more than one occasion over one million parts have been produced from one die set.

This carrier for the return spring of a choke control has a small cored hole, 1·2 mm diameter by 11 mm long, in its base. Two components are formed at each shot



Pressure die-cast components of a complete carburettor assembly



Recent Publications

Brief Reviews of Current Technical Books

Motor Cars Today

By H. E. Milburn, M.I.Mech.E.
London: OXFORD UNIVERSITY PRESS, Amen House, E.C.4. 1956. 8¾ × 5½. 295 pp. Price 15s.

In his foreword, the author states: "The purpose of this book is to reveal the secrets of the construction of a motor car, which is, to so many, a sealed box of mystery." This, of course, indicates that the work is intended mainly for owner-drivers and for younger readers who may be considering entering the industry. Nevertheless, the work is well illustrated with up-to-date examples of motor vehicles and components, and it deals with the subject of car design in a manner far more comprehensive than is usual in this type of book. In fact, it is technically sound and can be thoroughly recommended as an introduction to motor vehicle design.

The chapter headings are as follows: Historical and introductory; A general picture; The engine; The petrol system and the carburettor; Diesel engines; Electricity in the service of the motor car; Transmission; The final drive; Suspension; Steering; Brakes; Frames and bodies; How fast and how far; and Valediction.

That the work is comprehensive can be seen from the sub-headings of some of the chapters. For example, in Chapter 3, the following are discussed: Engine construction; Connecting rods, pistons and rings; Bearings; Valve arrangements; Lubrication; Cooling; Other essentials; The engine assembly; Two-stroke engines; Engine position; and Engine facts and figures. In Chapter 7, the items dealt with are: The clutch; Fluid flywheel; The gearbox; Easy gear changing; Preselector gearboxes; Hydraulic torque converters; Overdrive gears; Free-wheels; Automatic transmissions; Borg-Warner transmission; Hobbs transmission; Citroën hydraulic system; and Looking forward. Another interesting section is Chapter 12, in which the following are covered: The chassis frame; Unitary construction; The individual body; Body form; Interior appointments; External finish; Ventilation; and Driving comfort.

Trade Union Law

By H. Samuels, M.A.
London: STEVENS AND SONS LTD., 119-120 Chancery Lane, W.C.2. 1956. 8¾ × 5½. 95 pp. Price 12s. 6d.

The object of this, the fifth edition of Trade Union Law, is to present briefly an account of this branch of the law as it stands at present. Since the provisions of the Trade Disputes and Trade Unions Act, 1927, have now a purely historical interest, reference to them is made only in the appendix, in which the effect of the repealing Act of 1946, The Trade Unions and Trade Disputes Act, is set out; from this it can be seen that the rules pertaining to the Political Fund had to be amended to conform with the Trade Unions Act, 1913.

Subjects dealt with include the activities of Trade Unions in relation to the implication of the Law of Conspiracy, interference with contract, and damages for wrongful expulsion. An important feature is the supersession in 1951 of the Conditions of Employment and National Arbitration Order (S.R. and O., 1940/1305) by the Industrial Disputes Order (S.I. 1951/1376); this has materially affected the right to strike.

The chapters, or table of cases, are headed: Trade Unions in general, definition and classification; Membership of Trade Unions; Contracts of Trade Unions; Civil wrongs outside trade disputes; Civil wrongs where there is a trade dispute; Criminal conspiracy and intimidation; The rules of Trade Unions; The Political Fund; Registration of Trade Unions; Property and liabilities of Trade Unions—amalgamations—dissolution; and Legal proceedings. There are two appendices, one has already been mentioned, and the other gives the forms relating to Trade Union law.

The Properties of Aluminium and Its Alloys

London: THE ALUMINIUM DEVELOPMENT ASSOCIATION, 33 Grosvenor Street, W.1. 1955. 8¾ × 5½. 204 pp. Price 7s. 6d.

This is the A.D.A. Information Bulletin No. 2, first published in 1942. It was revised in 1944 and 1945. The latest edition includes the 1955 revision of the British Standards for Aluminium and Aluminium Alloys for General Engineering Purposes. It incorporates two substantial additions: the first includes data for alloys specified for aircraft use, and the second is tables of proprietary names under which many compositions are supplied. Among these tables are supplementary lists indicating the compositions of selected alloys produced overseas. Metric equivalents are now shown in the tables of properties.

The work is intended for use in conjunction with other Information Bulletins, among which is No. 1, which is a broad survey of the aluminium field, ranging from the distribution of aluminium ores in the earth's crust, and their reduction to metal, to the many applications of this metal. Bulletins 3-20 cover heat-treatment, joining, forming, handling and storage, extrusions, castings and finishing processes.

Among the contents are: Index of British Standards and aircraft specifications; The properties of aluminium and its alloys; British Standards for aluminium and aluminium alloys; Physical properties of aluminium alloys; Compositions and mechanical properties; Notes on mechanical test requirements and test methods; Super-purity alloys; Foil; Aluminium for electrical conductors; Clad materials; Other special materials; Notes on characteristics affecting design; General guides to alloy selection. There is also an appendix, containing proprietary names, as well as a general index.

Automation in Theory and Practice

Oxford: BASIL BLACKWELL, 49 Broad Street. 1956. 7½ × 5. 140 pp. Price 12s. 6d.

In the Michaelmas Term, 1955, a series of lectures was given at Oxford to examine automation, both from the technical viewpoint and in respect of its social and economic significance. Since no book covering this field was available, it was decided to preserve these lectures in more permanent form. This has been done in the work under review. Although the scripts have been edited by the lecturers, alterations have generally been limited to such expansions or contractions as were suggested by the discussions that followed each lecture, or to those minor changes needed to turn a verbatim report into continuous prose. In general, it can be said that the book will be of considerable interest to those who are concerned with automation and its implications in the broadest sense.

The lecturers include the Rt. Hon. The Earl of Halsbury, F.R.I.C., F.Inst.P., Managing Director of the National Research Development Corporation; R. H. Macmillan, M.A., Mem.A.S.M.E., Member of the Society of Instrument Technology, Lecturer in Engineering, University of Cambridge; Frank G. Woollard, M.B.E., M.I.Mech.E., M.I.Prod.E., M.I.I.A., M.S.A.E., consulting engineer; H. R. Nicholas, O.B.E., National Secretary, Metal and Engineering Group, Transport and General Workers' Union; W. R. Spencer, A.S.A.A., F.C.W.A., M.I.I.A., Director of Urwick, Orr and Partners Ltd., management consultants; Michael Argyle, M.A., Lecturer in Social Psychology, University of Oxford, also Director of Studies, Cambridge Managers' Course, 1954-1955, and in charge of part of a D.S.I.R. research project into social factors in productivity; E. M. Hugh-Jones, M.A., Fellow and Tutor of Keble College, Oxford, Lecturer in Economics.

After the introduction, there are six chapters, two appendices, a section headed Further reading, and an index. The chapters are entitled: The scientific basis; Automation in engineering production; The trade union approach to automation; Administrative applications of automation; Social aspects of automation; and Automation to-day. In Appendix I, typical work of an administrative office is outlined, and Appendix II gives a simplified description of a computer.

Speedicut Manual of Screw Thread Tools

Sheffield: FIRTH BROWN TOOLS LTD., Speedicut Works, Carlisle Street East, Sheffield 4. 1956. 7½ × 5. 321 pp. Price 25s.

In preparing this manual, the aim of the publishers has been at providing an informative and practical guide to promote greater understanding of screw thread tools and techniques among those whose technical knowledge of this subject is limited. The work is also intended to serve as a useful reference for more experienced engineers. Whilst various methods of thread production are dealt with, the emphasis is mainly on screwing-taps. Up-to-date screw-tap nomenclature and definitions are given, and many different types of taps in general use are described. Factors involved in their design and maintenance are explained. The information is such as to furnish guidance in the selection of the most suitable kind of tool for a wide range of materials. Comprehensive lists are given of tapping drill sizes and speeds, together with notes on lubrication. Causes and remedies of tap faults and failures in operation are also discussed. There are notes on the tapping of specific materials, and these include information regarding plastics and hard rubber. In short, the book is most useful for reference purposes in the workshop.

The Fatigue of Metals

London: THE INSTITUTION OF METALLURGISTS, 28 Victoria Street, S.W.1. 1956. $8\frac{3}{4} \times 5\frac{1}{2}$. 164 pp. Price 25s.

This book comprises a series of five lectures given at Llandudno in 1955. The lectures were concerned with the general problems of the fatigue of metals, and they were followed by discussions. However, in order to restrict the length of the volume, records of the discussions are not included.

The book is an authoritative treatise, such as one would expect from this Institution. Fundamental considerations on the fatigue of metals are dealt with by J. Holden, M.Sc., Ph.D., in the first chapter. The second chapter, which is by G. Forest, B.Sc., A.M.I.Mech.E., A.F.R.Ae.S., is entitled "The effect of fatigue of notches, surface finishings, etc." Then the structural aspects of aircraft fatigue are covered by P. B. Walker, C.B.E., M.A., Ph.D., A.I.M. Chapter 5 is on the effect of temperature on fatigue properties, by P. H. Frith, A.Met., F.I.M. The work is of considerable interest and value, and includes most useful lists of references.

Automobile Engines

By Arthur Judge, A.R.C.Sc., D.I.C., Wh.Sc., A.M.I.A.E.
London: CHAPMAN AND HALL LTD., 37 Essex Street, W.C.2. 1956. $7\frac{1}{2} \times 5$. 474 pp. Price 21s.

It is now some 30 years since this well-known book was first published. This is the sixth edition, the previous one having been fairly extensively revised. The new information that has been included is mainly on high octane motor fuels, pre-ignition and post-ignition, and allied subjects: post-ignition is the term that this author gives to the phenomenon commonly called running-on. Several new fuels for normal and high output engines are described and their properties considered. More space has now been devoted to the subject of combustion chamber design and car engine performance.

The section dealing with engine components has been extensively revised and new data and illustrations incorporated. Among the subjects dealt with more fully are the sections on valves and their actuating gear, valve timing diagrams, pistons, bearings, crankshafts, engine mountings and torsional vibration dampers. The sections on engine cooling and lubrication have also been revised and extended. Information and illustrations on the new small, high-speed diesel engines is also included. Another new chapter deals with automobile gas turbines. In this chapter, both the theoretical and practical aspects of these power units are considered: the more promising working cycles and types are dealt with, and typical engines are described. Some interesting information is given on special engine types, such as swash-plate units, and sleeve valve and two-cycle engines. The book was written originally for the motor engineer, mechanic and student, so the treatment is essentially of an elementary and largely practical nature.

Automobile Wiring Diagrams

By E. Molloy.
London: GEORGE NEWNES LTD., Tower House, Southampton Street, W.C.2. 1956. $8\frac{3}{4} \times 5\frac{1}{2}$. 311 pp. Price 30s.

The wiring diagrams given in this book cover the majority of British cars from 1938 onwards. A selection of Continental car diagrams have also been included. In all, the total number of models dealt with is over 500. An introductory section describes the Lucas cable colour coding system, how to trace out a car wiring diagram, and fault location procedures for the charging circuit; it also deals with the ignition circuit, the starter motor circuit, and lighting and accessory circuits. In addition, three methods of carrying out wiring renewals are described. This whole introductory section covers the first 20 pages. The remainder of the book is entirely devoted to wiring diagrams, except for the last five pages, on which there is an index.

INSTITUTION OF MECHANICAL ENGINEERS

Forthcoming Meetings of the Automobile Division

MARCH

Birmingham

Tuesday, 26th March, 6.30 p.m., in the James Watt Memorial Institute, Great Charles Street, Birmingham. Paper: A Review of the Hydrokinetic Fluid Drives and Their Possibilities for the British Motor Industry," by J. G. Giles (Associate Member).

Derby

Monday, 18th March, 6.15 p.m., in the Rolls-Royce Welfare Hall, Nightingale Road, Derby. Joint Meeting with the Derby Branch of the Royal Aeronautical Society. Paper: "Some Operating Problems of Jet Air Liners," by W. O. W. Challier, Dipl. Ing., F.R.Ae.S.

North-Eastern

Wednesday, 20th March, 7.30 p.m., in the Chemistry Lecture Theatre, The University, Leeds. Paper: "Transmission Developments for Public Service and Heavy Goods Vehicles," by A. Gordon Wilson (Associate Member).

North-Western

Tuesday, 19th March, 7.15 p.m., in the Engineer's Club, Manchester. Address by the Chairman of the Automobile Division, A. G. Booth, M.B.E. (Member), entitled "Experiences During Forty Years of Automobile Design."

Scottish

Monday, 18th March, 7.30 p.m., in the Institute of Engineers and Shipbuilders, 39, Elmbank Crescent, Glasgow, C.2. Paper: "Rubber Springs for Vehicle Suspension," by A. E. Moulton, M.A., and P. W. Turner, M.A., B.Sc.

Western

Thursday, 28th March, 6.45 p.m., in the Royal Hotel, Bristol. Paper: A Review of Hydrokinetic Fluid Drives and Their Possibilities for the British Motor Industry," by J. G. Giles (Associate Member).

APRIL

London

Tuesday, 9th April, 4.30 p.m., at 1 Birdcage Walk, Westminster, S.W.1. Automobile Division General Meeting. Symposium on Superchargers and Supercharging:

1. *"The Turbo-charging of High Speed Diesel Engines—Present Position and Future Prospects," by C. H. Bradbury (Member).*
2. *"Design and Development of Small Radial-flow Turbochargers," by C. A. Judson, B.Sc. (Associate Member) and E. Kellett.*
3. *"The Supercharging of High-speed Diesel Engines by Mechanically Driven Compressors," by B. W. Millington, B.Sc. (Eng.), (Associate Member).*
4. *"An Approach to the Problem of Pressure Charging the Compression Ignition Engine," by D. W. Tryhorn, B.Sc. (Associate Member).*

North-Western

Tuesday, 9th April, 7.15 p.m., at Leyland Motors Ltd., Leyland, Lancs. Paper: "Transmission Developments for Public Service and Heavy Goods Vehicles," by A. Gordon Wilson (Associate Member).

Western

Thursday, 25th April, 6.45 p.m., in the Royal Hotel, Bristol. Paper: "Sales and Service in the Transport Industry," by J. M. Forbes.

HEENAN & FROUDE G.4 DYNAMOMETER

A New Fluid-friction Machine that is Compact, Simply Controlled, and Reversible to Extend its Operating Range

It is difficult to give consideration to any period in the history of the internal combustion engine without encountering the ubiquitous Froude fluid-friction, absorption dynamometer. It has been used for the testing and development of all types of power unit, from the smallest motorcycle engine to the gigantic marine engine. Figuratively, it has nursed the internal combustion engine through infancy, assisted it in adolescence, and developed its latent possibilities in maturity. Throughout this long period of useful activity it has remained basically unchanged, although it has been continually subjected to detail improvement in design, construction, materials and performance. In view of this record, the advent of an entirely new model by Heenan and Froude Ltd., of Worcester, is of considerable interest. It is designated the G-type to differentiate it from the earlier S.G.- and D.P.-types. Prototype machines, which have undergone exhaustive testing at the maker's works and are in service at certain plants represent the culmination of two years' intensive development work. Commercial production will be commenced in the near future, but at present the machines are not generally available.

While it retains the same basic principle of operation, it will be seen that the design is entirely new in conception. The prototype model has a power absorption capacity of 350 h.p. at a maximum rotational speed of 6,000 r.p.m. Other models to absorb much higher powers and also to operate at much higher speeds are undergoing development.

Occupying a floor space approximately 3 ft × 2ft, the new G.4 dynamometer has a power absorption capacity of 350 h.p. at speeds up to 6,000 r.p.m.

The main advantages of the new machine may be listed as:

1. An outstanding feature is its compactness. The overall dimensions of the 350 h.p. machine are only $37\frac{1}{4}$ in $\times 25\frac{3}{4}$ in $\times 38$ in high.
2. To simplify operation, the long-familiar sluice-gate method of control has been discarded. Instead, load regulation is effected by a small hand wheel on a plunger valve controlling admission of water to the vaned rotor. The control can readily be motorized, if required.
3. A load throw-off device is incorporated as standard. This facilitates tests on governed engines.
4. Contact seals and glands have been eliminated, thereby securing several advantages. The shaft bearing centres are shortened and whirling speeds are raised to much higher values. The need for maintenance and consequent idle time is reduced. An indirect advantage is that the static balance of the dynamometer can be checked without the necessity of disconnecting it from the engine undergoing test.
5. Provision is made for the addition, as standard accessory equipment, of a friction brake to reinforce the power absorption capacity at low rotational speeds.
6. An alternative, and lower, power absorption range can be obtained automatically by turning the dynamometer through 180 deg or setting up the engine for test on the opposite side of the machine. In other words, there are two power ranges according to whether operation is at normal forward rotation or at reverse rotation.
7. The height-adjusting gear on the spring balance has been eliminated.
8. Weights used to supplement the spring balance are small and easily handled, weighing 10 lb each.
9. Load is unaffected by inlet valve setting.
10. A propeller-law characteristic is maintained over the entire operating range.
11. Even a low pressure of inlet water supply is adequate to operate the dynamometer throughout its range.
12. Total weight is only 6 cwt, that is, only 40 per cent of the current DPX.4 dynamometer of similar capacity.

The rotor and stators, of cast nickel bronze or aluminium bronze, are of the long-established Froude design and are housed in a circular carcass. This unit is trunnion mounted in conventional ball-bearings in the horizontally divided outer casing of cast aluminium. Axial location is secured on one bearing only and the carcass is sealed by wet and dry labyrinth seals at each end. Any leakage of water past the seals is drained to the main sump in the base of the outer casing.

Mounted on the dynamometer shaft is a simple centrifugal pump which draws water from the sump and delivers it, past the control valve and through passages in the stator vanes, to the vortex chambers. This constitutes a major change from earlier designs in which the water supply was delivered directly to the vortex chambers. The effect is that the water feed is adjusted to the speed and any

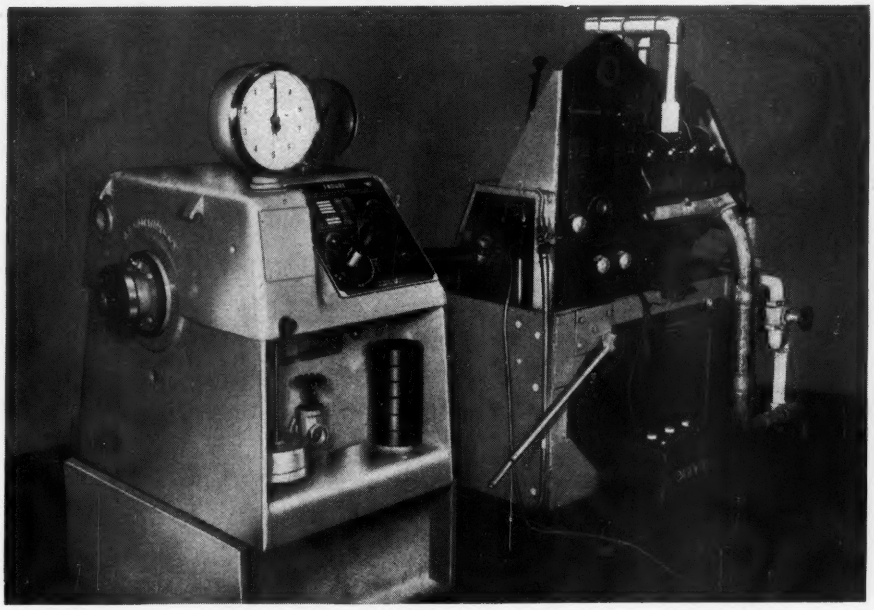
fluctuations of the supply pressure, delivery, or temperature are rendered innocuous.

Should the dynamometer speed tend to rise, more water is automatically pumped into the rotor vortex chambers, thus increasing the resistance of the dynamometer and checking the rise of rotational speed. Conversely, should the speed tend to fall, the resistance is lessened and the speed is thereby maintained. By this means it is ensured that, for any particular setting of the control valve, a propeller-law characteristics (h.p. $\propto N^{2.8}$) is established and stable operating conditions are obtained. In order to change the load or the speed, it is necessary merely to adjust the setting of the control valve.

Water supply is admitted through an inlet valve to the usual spring balance mounted on the top of the outer casing. Although the lever has a ratio of 40:1, approximately three times that of the 5 ft lever on the DP-type dynamometer, it is housed completely within the outer casing. The range of the balance is extended by the use of small deadweights of 10 lb, to B.O.T. limits, instead of the usual 50 lb weights previously employed. From the illustrations it will be noted that the lever weight bolt and also the stack of weights are accommodated within the pedestal. By virtue of the high ratio of the lever, the range of movement is so small that the need for height-adjusting gear on the spring balance disappears.

A dashpot of the fluid-transfer type is included to damp any oscillation of the balance and is adjustable by means of

The new dynamometer with a Ford V.8 engine on test in the Heenan and Froude research department



sump in which a level is maintained by two weirs of different heights. From the sump it is lifted to the vortex chambers of the rotor by the pump and, after being heated in operation, is returned at high velocity to the sump by way of a fixed-area orifice in the dynamometer carcass and a stand pipe discharging between the two weirs. In this way, turbulence is avoided; the hot water flowing over the low weir into the main sump where it is mixed with and cooled by the inlet water. The warmer water overflows the high weir and is discharged to waste. In the event of an interruption of the water supply, no immediate emergency will arise as water from the level in the sump will continue to circulate although, of course, its temperature will rise.

The temperature of the water leaving the dynamometer carcass is indicated on a thermometer mounted on the control panel. By regulation of the inlet valve so as to maintain a water outlet temperature of approximately 140 deg F, the utmost economy of water is attained.

In passing, it is of interest to recall that this method of indirect delivery of water by means of a dynamometer-driven pump was first used in a dynamometer hastily evolved to test the special 3,300 h.p. Rolls-Royce R-type engine which powered the successful Supermarine S 6 B seaplane in the 1931 Schneider Cup Race.

Torque measurement is by means of a simple reducing lever system connecting the dynamometer carcass to the a small needle valve. The linkage of spring balance, dashpot, and weight bolt to the lever is effected by p.t.f.e. bushes on a stainless steel spindle. Speed indication is conventional. The tachometer, mounted on the top of the casing, may be of mechanical, electrical, or pulse type, as desired. In the horse-power formula for the dynamometer,

$$\text{b.h.p.} = \frac{W \times N}{F}$$

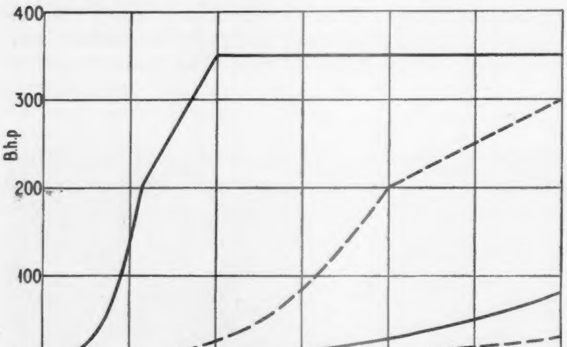
the factor F for this machine is 200.

When governed engines are on test, it is of advantage to have means of throwing off the load quickly in order that the behaviour of the governor can be assessed. Incorporated in the main control valve is a ported sleeve controlled by a lever. Rotation of this sleeve through 90 deg cuts off the supply of water from the pump to the vortex chambers and the dynamometer carcass immediately empties and the load is thrown off. The setting of the load control is not affected and load is restored by merely reversing the operation on the throw-off lever. Inadvertent operation of the throw-off lever is prevented by a plunger-type detent operated by a small lever located in a shielded position in the pedestal immediately below the control panel.

The main control valve is of the plunger type, actuated by a hand wheel provided with a graduated sleeve for setting purposes. Load is taken up by turning the hand-wheel to advance the valve plunger, thus commencing to

open the port by way of which water is delivered by the pump to the vortex chambers and, simultaneously, commencing to close the by-pass from the dynamometer carcass discharge. Maximum load is imposed when the admission port is fully open and the by-pass is fully closed. The object of this arrangement is to give an approximate straight-line control at constant speed.

The capacity of the G.4 machine is shown in the power absorption curves, the capacity in reverse rotation being indicated in broken line. This will be the first size of the



Power absorption curves for the G.4 dynamometer operated in either forward or reverse rotation

new type to be marketed. The fitting of a friction brake will, in both the forward and reverse capacities, flatten the maximum curve in the lower speed range, giving a closer approach to straight-line performance and more accurate assessment of the engine output in the lower speed ranges.

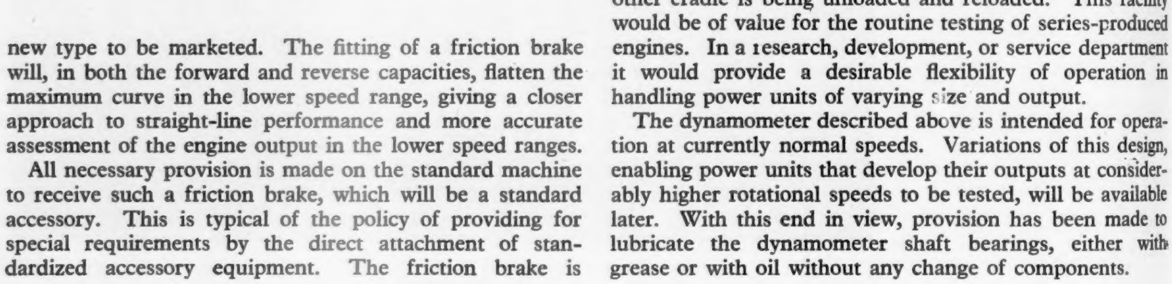
All necessary provision is made on the standard machine to receive such a friction brake, which will be a standard accessory. This is typical of the policy of providing for special requirements by the direct attachment of standardized accessory equipment. The friction brake is currently undergoing testing and development, and most probably will not be immediately available when the dynamometer is put into production. Remote motorized control can be provided where required. This is of value when tests of long duration are to be undertaken and also when the machine is used for sequenced programme tests. Whereas a motor of from ½ h.p. to ¾ h.p. was necessary for the operation of the control sluice gates or diaphragm plates of earlier types, a miniature motor of about 1/100 h.p. is adequate for the control of the new model.

In the general design of the machine full advantage has been taken of the ability to operate in either direction of rotation. It is, in effect, two dynamometers of different, but overlapping, capacities combined in a single machine. Where only a single engine cradle is provided, the dynamometer can be lifted by a crane, turned through 180deg, and lowered into position again. Holding-down bolts are symmetrically arranged in plan about a central point coincident with the intersection of vertical planes respectively containing the axis of the dynamometer shaft, and normal to that axis and located midway between the shaft half-couplings. Lifting hooks are cast on the upper half of the casing. Alternatively, the dynamometer can be mounted on a sole plate having a centre-point spigot which is engaged in a circular recess provided, as standard, in the machine base. With this mounting, it is merely necessary to withdraw the four fixing bolts, swivel the dynamometer on the sole plate, and replace the bolts.

The provision of an engine cradle at each side of the machine enables greater utilization of the dynamometer to be made, as an engine can be tested on one side while the other cradle is being unloaded and reloaded. This facility would be of value for the routine testing of series-produced engines. In a research, development, or service department it would provide a desirable flexibility of operation in handling power units of varying size and output.

The dynamometer described above is intended for operation at currently normal speeds. Variations of this design, enabling power units that develop their outputs at considerably higher rotational speeds to be tested, will be available later. With this end in view, provision has been made to lubricate the dynamometer shaft bearings, either with grease or with oil without any change of components.

General arrangement of the dynamometer. Shaft half-couplings and holding-down bolts are symmetrical about a common central point to permit turning through 180 deg to reverse operational rotation



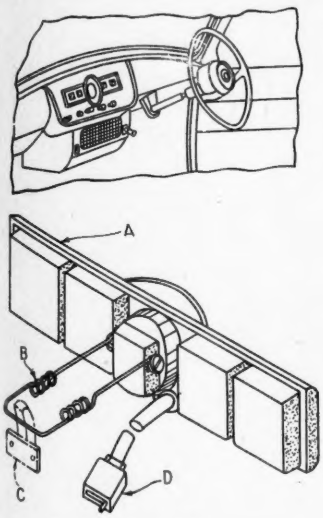
CURRENT PATENTS

A REVIEW OF RECENT AUTOMOBILE SPECIFICATIONS

Group assembly of electrical instruments

Assembly is facilitated by this method of grouping instruments to form a single unit which is inserted in the facia board and connected by cable harness furnished with a multi-point plug. Preferably only one terminal of each instrument is connected to the plug, the other terminal being earthed to the vehicle frame. The group is, of course, assembled on the bench, ready to be supplied to the vehicle assembly line.

The common housing A accommodates a centrally disposed speedometer, a cooling water thermometer, gauges for fuel and for oil pressure, and a combination light comprising tell-tale lamps associated with headlights, direction indicators,



No. 751690

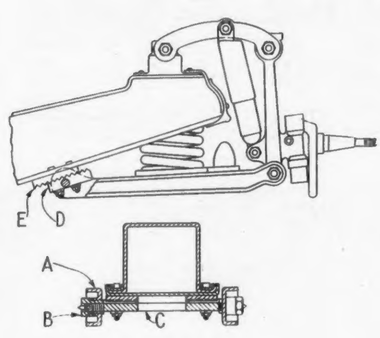
starter, or the like. To secure the housing in position in an aperture in the facia board, a bowed tension spring B is employed. This is simply engaged over a suitable hook C fixed to the dashboard or other convenient member of the vehicle framing.

All the instruments and lamps are wired to a single cable terminating in the multi-point plug D by which connection is established with a complementary socket mounted on the dashboard. *Patent No.* 751690. *Daimler Benz A.G. (Germany).*

Adjustment of camber angle

On a wishbone-type independent wheel suspension, adjustment of the wheel camber angle is arranged by means of a displaceable attachment of the lower arm pivot to the framing. The provision of such a device obviates the common practice of bending the arm.

In the example illustrated, the inner ends of the lower arms A are threaded for the reception of threaded bushing nuts B screwed on the ends of spindle C.



No. 751780

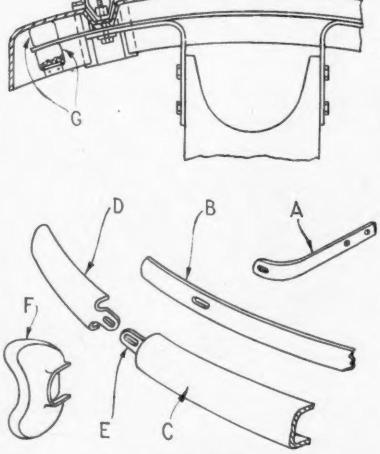
Secured to, or integral with, the spindle are a pair of transversely serrated locking plates D. Complementary anchor plates E are welded or bolted to the vehicle frame.

The securing bolts, which may serve to locate the anchor plates on the frame, pass through slotted holes in the locking plates. To adjust the camber the nuts are loosened so that the serrations are disengaged and then, after moving the wheel to its new alignment, again drawing up the nuts to secure. *Patent No.* 751780. *L. J. Carpezzi (U.S.A.).*

Bumper construction

In this composite structure, a shock-absorber bar is attached to two frame-mounted brackets and is enclosed on three sides by a protective shell and overriders. Only two bolts are required to make the assembly and attach it to the frame brackets.

Brackets A are bolted to the vehicle framing and support the bowed shock-absorber bar B. The shell is formed in three parts; a mid-portion C and two end portions D. Both ends of part C and the inner ends of parts D are furnished with depressed slotted lugs E which overlap



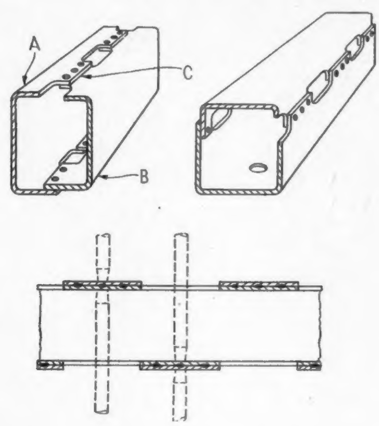
No. 750382

and are aligned with slots in bar B and brackets A. Overriders F are provided with a yoke which seats over the lugged ends of the shell parts and carries a welded-on nut. On each side of the vehicle, a bolt inserted through parts A, B, C and D is screwed into the nut on the overrider to draw up the assembly.

The end parts D of the shell may be resiliently supported from the bar B by interposed rubber blocks G, the rearward one seating on a strap member welded across the shell at a suitable position. *Patent No.* 750382. *Daimler-Benz A.G. (Germany).*

Box-section frame members

This box-section structural member, a chassis side frame, for example, is formed of two channel sections arranged in overlapping connection. The object of the invention is to enable the assembly to be



No. 752036

secured by spot-welding by means of simple tools and to obtain a favourable strength-to-weight ratio.

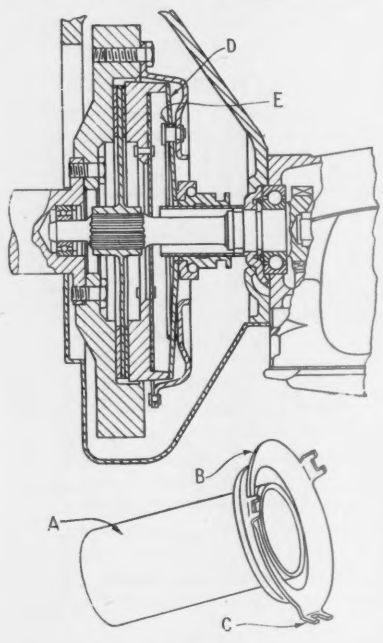
Each side frame consists of an outer channel A and an inner channel B, the webs of both channels being scalloped to leave marginal projections C. An essential feature is that in each channel element the scallops in one web are offset to those in the other web. When assembled with the projections overlapping, the resulting box-section is apertured by elongated holes in both upper and lower runs. Since the scallops are offset in each element, the apertures in one run give access by simple tools to the projections on the other run.

Shown in the diagram is the set-up for spot welding the section on an automatic welding machine. Each aperture is entered simultaneously by sets of electrodes corresponding to the number of welds to be made. Alternatively, a single set of electrodes may be used in each aperture and the plurality of welds be made by "shuffling" either the electrodes or the work. *Patent No.* 752036. *The Budd Company (U.S.A.).*

Clutch construction

Economy in material, number of parts, and assembly time in mass production is claimed for this clutch construction; in particular, for the guide sleeve upon which the throw-out ring operates. Commonly, the sleeve is flanged, registered, and bolted to the wall of the clutch casing or the gear unit. In this instance, it is resiliently clamped between the clutch casing and the attached gearbox.

A sketch of the guide sleeve A on an enlarged scale is given in the illustration. To the sleeve proper is welded, brazed, or soldered a stepped flange B stamped from sheet metal of relatively thicker section. The flange is formed at the periphery with three forked tongues C inclined at 45 deg to the axis. The flange is centred and axially located in the aperture in the clutch housing, with the



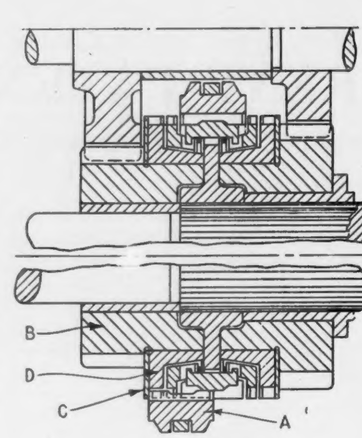
No. 752059

three tongues accommodated in suitably formed recesses. These tongues project beyond the face of the housing and are resiliently deformed when the gearbox is bolted in position.

It will be noted that the clutch has a disc-shaped spring D rocking about pivot rings E. *Patent No.* 752059. *Adam Opel A.G. (Germany).*

Synchromesh gears

The invention relates to synchromesh gears of the type in which the toothed wheel is combined with a pressed-on or shrunk-on toothed clutch member for engagement by the gearshift sleeve. This construction is resorted to in order to reduce the length, and consequently the weight, of the gearing assembly. A further means of saving length is to shorten the length of the gearshift sleeve, but this expedient has the disadvantage that it largely exposes the stops mounted on the gearshift member when a gear is engaged. Under the influence of centrifugal force the stops show a tendency to tilt which, under certain conditions, may cause the sleeve to swing. As a result the sleeve



No. 751698

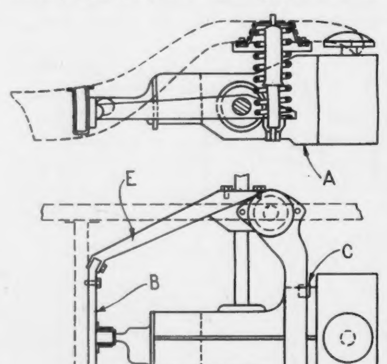
tends to become more deeply engaged in the gear and the shifting fork is subjected to increased wear.

To avoid this drawback while retaining the advantages of a narrow gearshift sleeve A, the gear B is provided with a stop disc C located between the gear and the clutch member D. When a gear is engaged, as shown in the lower part of the drawing, a sleeve of minimum length is displaced until one edge encounters the stop disc and the other is approximate to the vertical axis of the stop. Any tendency for the stops to tilt produces a thrust on the sleeve which is taken up by the stop disc, and swinging and excess travel is prevented. *Patent No.* 751698. *Volkswagenwerk G.m.b.H. (Germany).*

Suspension of driven wheels

To meet the requirements of conveyor belt assembly of motor vehicles, the power unit, driving gear with swinging half-axles, and an independent wheel suspension system, together form a sub-assembly that can be attached to the vehicle frame by six bolts. In the drawing the method is applied to the driven rear wheels of a vehicle, but it can be used also for driven, dirigible front wheels.

The driving unit A comprises engine, clutch, gearbox, and axle gear. At its



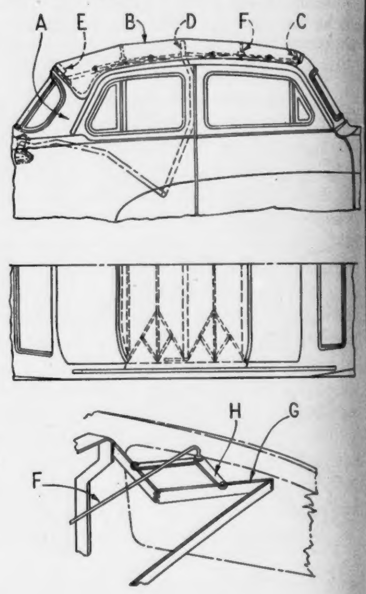
No. 752041

forward end it is attached by a resilient rubber mounting to transverse member B and two further rubber mountings C are provided between the clutch housing and a cross member D. Wheel arms E are pivoted obliquely to the ends of member B and at their free ends, beyond the axle bearing, are extended laterally and dished to receive the helical suspension springs. The upper ends of the springs are seated in depressions in member D and mounted within the springs are shock absorbers.

On the vehicle, transverse member B is secured by two bolts to a chassis cross member, and cross member D by two bolts in way of the spring supports to each of the chassis side frames. *Patent No.* 752041. *Volkswagenwerk G.m.b.H. (Germany).*

Convertible construction

In the conventional convertible vehicle with a collapsible hood affording the roof and the upper rear wall, the rear of the



No. 752282

body lacks the rigidity obtained with the saloon body. As a consequence, additional lateral bracing is necessitated, increasing production costs. The invention is characterized by an upper rear wall that is a permanently fixed part of the body structure. A top opening is defined by the upper margins of the windscreen frame, the two side walls, and the rear wall.

The example shows such a body, having a rigid rear wall A. A fabric hood B is carried by a horizontally disposed and rectangular frame that can be collapsed in its own plane. The frame comprises front, central and rear main struts, C, D and E respectively, and intermediate auxiliary struts F. On each side, the ends of the main struts are linked by folding side members G. Struts F are connected, by further folding links H, to members G. The central main frame strut D is pivotally mounted in the body in such manner that the frame and hood, when collapsed, can be swung down bodily past the forward face of the rear wall and into the stowed position. *Patent No.* 752282. *Austin Motor Co. Ltd.*

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